FalconSat-2

Structural Verification Report

Appendix A
Derivation of Limit Loads

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May 13, 2002

Background:

The FS2 development-test program (testing of the Engineering Model, EM) provided data that was of use for detail design. Response accelerations measured in the random-vibration test indicate mode shape and frequency for key modes of vibration, as well as acceleration levels.

Of key importance, the results of EM random-vibration testing indicate that this test environment will stress the FS2 primary structure considerably more than the preliminary design loads derived from the quasi-static load factors specified in CARS (see Appendix A of the FalconSat-2 SVP). The flight structure was designed for the accelerations and corresponding mode shapes measured in EM testing.

For qualification and acceptance testing (testing of the Qualification Model, QM, and Flight Model, FM, respectively), the specified random vibration environment was notched in the X and Y (lateral) directions near the fundamental lateral frequency to protect the hardware from unrealistically high loads. (Appendix D justifies this notching.) As a result, measured accelerations associated with this mode were lower than for EM testing. The following analysis derives "design" loads from the QM test data and data reduction documented in Appendix E. These loads are used for the final strength analysis (Appendix B) and for fracture mechanics analysis (FalconSat-2 Fracture Control Compliance Report).

Problem Statement:

From the QM test data, derive final limit loads for analytically verifying structural requirements for the FS2 structure. Loads will be derived for the primary box structure, the bolted interface to the separation ring, the bolted interface between the equipment column and the base plae, and the bolted interface between the battery and the baseplate. All of these loads will be used to assess the baseplate, which is the critical structural part, along with the bolts attaching to the baseplate.

Process:

1. From accelerations measured at different locations on the QM and the finite-element model documented in Appendix C, identify the approximate mode shapes for the key modes of vibration.

2. Estimate mass, center of gravity, and moments of inertia for the FS2 box assembly, the column (with equipment), and the battery pack. (Basis: Appendix G--Mass Properties Report, Version 8.0)

3. From the identified mode shapes, derive sets of limit rigid-body translational and rotational accelerations applicable to the box, the column, and the battery pack for qualification testing.

4. From the mass properties and the rigid-body accelerations, calculate limit forces and overturning moment for qualification testing at the key interfaces. Also, compute limit loads for flight, which are the qualification loads divided by 1.4 to reflect the 3-db qual margin.

Step 1. Identify Key Mode Shapes Determined in Qualification Testing

The fundamental axial mode is at 310 Hz, and another axial mode is at 480 Hz. Figures E-3 and E-4 from Appendix E show the response peaks at these frequencies. The finite-element model FS2QM_4, which was correlated for the natural frequency of the fundamental lateral mode, predicts two axial modes, one at 297 Hz and the other at 514 Hz. From the Nastran plots, most of the mass of the outer box structure participates for both of these modes, with little contribution of the interior components. The RMS acceleration for these two modes combined is about 35 g, as noted in Sec. E.2a of Appendix E.

The FS2QM_4 Nastran model predicts two lateral modes, both of which are best characterized as rocking modes of the exterior box structure. The mass of the column, counting that of the mounted equipment, does not move as a rigid body with the box. This makes sense because most of the strain energy for this mode is in bending of the baseplate ribs outside the separation ring. These modes can be closely approximated by combining lateral and rotational rigid-body accelerations applied to the box mass only. The FS2QM_4 model was correlated with qualification test results to predict about the same fundamental frequency (154 Hz in test and 153 Hz for the model). According to the model, this mode consists of X-axis acceleration only. The Y-axis mode is predicted by the model to be at 156.5 Hz. Test results show the modal frequencies to be 157 Hz and 171 Hz. Both modes appear to contain both X and Y motion, with the 157-Hz mode being more X than Y and the

The column itself appears from Fig. E-9 to have rocked in a fundamental frequency of 500 Hz, although the mode was not clearly pronounced in this figure.

The battery responded highly in two rocking modes. For X motion, the natural frequency was 300 Hz, and, in Y, it was 370 Hz.

To calculate the limit shear and moment at the FS2/ring interface, the column/base-plate interface, and the battery/base-plate interface, first compute separately the mass and moments of inertia for the box, the column (with equipment), and the battery.

Step 2a. Mass Properties for FS2 Box (external structrure plus mounted equipment)

From Appendix G: Weight of base plate (lb), $Wbp := 4.49$

Weight of top plate assembly, $Wtp := 4.72$

Weight of side panel assemblies, $Wsp := 14.32$

Total box weight (lb), $Wbox := Wbp + Wtp + Wsp$ $\quad Wbox = 23.53$

Given: Box dimensions (in): $xbox := 12.5$

$ybox := 12.5$ $\quad$ Thickness (in), $t := 0.75$

$zbox := 12.5$ $\quad$ Mass (lb.s^2/in), $Mbox := \dfrac{Wbox}{386.1}$

$$\boxed{Mbox = 0.0609}$$

Assume c.g. is located at geometric center of the box.

Z distance of box c.g. to critical interface (in), $Zbarbox := \dfrac{zbox}{2}$ $\quad Zbarbox = 6.25$

Compute volume (cubic inches):

Volume of solid box, $V1 := xbox \cdot ybox \cdot zbox$ $\quad V1 = 1.9531 \times 10^3$

Volume of hollow interior, $V2 := (xbox - 2 \cdot t) \cdot (ybox - 2 \cdot t) \cdot (zbox - 2 \cdot t)$

$V2 = 1.331 \times 10^3$

Volume of box wall, $V := V1 - V2$ $\quad V = 622.125$

Compute box masses (lb.s^2/in), assuming same density as for hollow box:

Mass of solid box, $M1 := Mbox \cdot \dfrac{V1}{V}$ $\quad M1 = 0.1913$

Mass of hollow interior, $M2 := Mbox \cdot \dfrac{V2}{V}$ $\quad M2 = 0.1304$

Compute moment of inertia (MOI) about centroidal X axis for the assumed solid and hollow boxes:

MOI for solid box (lb.s^2.in):

$$J1x := \frac{M1}{12} \cdot \left(ybox^2 + zbox^2\right) \qquad J1x = 4.9825$$

MOI for hollow interior (lb.s^2.in):

$$J2x := \frac{M2}{12} \cdot \left[(ybox - 2 \cdot t)^2 + (zbox - 2 \cdot t)^2\right] \qquad J2x = 2.6294$$

MOI for hollow box (lb.s^2.in):

$$Jboxx := J1x - J2x \qquad \boxed{Jboxx = 2.3531}$$

Step 2b. Mass Properties for the Column (with mounted equipment)

From Appendix G:

Column weight (lb), $Wcol := 3.18 + 2.39 + 2.77 + 2.12$ $Wcol = 10.46$

Given: Dimensions (in): $xcol := 7.50$ Thickness (in), $t := 0.75$

$ycol := 7.50$

$zcol := 8$ Mass ($lb.s^2/in$), $Mcol := \frac{Wcol}{386.1}$

$\boxed{Mcol = 0.0271}$

Assume c.g. is located at geometric center of the column.

Z distance of box c.g. to critical interface (in), $Zbarcol := \frac{zcol}{2}$ $Zbarcol = 4$

Compute volume (cubic inches):

Volume of solid column, $V1 := xcol \cdot ycol \cdot zcol$ $V1 = 450$

Volume of hollow interior, $V2 := (xcol - 2 \cdot t) \cdot (ycol - 2 \cdot t) \cdot (zcol - 2 \cdot t)$ $V2 = 234$

Volume of box wall, $V := V1 - V2$ $V = 216$

Compute column masses ($lb.s^2/in$), assuming same density as for hollow column:

Mass of solid column, $M1 := Mcol \cdot \frac{V1}{V}$ $M1 = 0.0564$

Volume of column wall, $V := V1 - V2$

Mass of hollow interior, $M2 := Mcol \cdot \frac{V2}{V}$ $M2 = 0.0293$

Compute moments of inertia (MOI) for the assumed solid and hollow columns:

MOI for solid column ($lb.s^2.in$):

$$J1x := \frac{M1}{12} \cdot \left(ycol^2 + zcol^2\right) \qquad J1x = 0.5656$$

MOI for hollow interior ($lb.s^2.in$):

$$J2x := \frac{M2}{12} \cdot \left[(ycol - 2 \cdot t)^2 + (zcol - 2 \cdot t)^2\right] \qquad J2x = 0.1914$$

MOI for hollow column ($lb.s^2.in$):

$$Jcolx := J1x - J2x \qquad \boxed{Jcolx = 0.3742}$$

Step 2c. Mass Properties for the Battery Pack

Given: Dimensions (in):

$xbat := 2.80$

$ybat := 4.80$

$zbat := 7.05$

From Appendix G:

Weight (lb), $Wbat := 4.32$

Mass (lb.s^2/in), $Mbat := \frac{Wbat}{386.1}$

$\boxed{Mbat = 0.0112}$

Assume c.g. is located at geometric center of the battery pack:

Z distance of box c.g. to critical interface (in), $Zbarbat := \frac{zbat}{2}$ $\quad Zbarbat = 3.525$

Compute volume (cubic inches):

Volume of solid box, $V := xbat \cdot ybat \cdot zbat$ $\quad V = 94.752$

Compute moment of inertia (MOI) for the battery pack:

MOI for solid box (lb.s^2.in):

$$Jbatx := \frac{Mbat}{12} \cdot \left(ybat^2 + zbat^2\right) \qquad \boxed{Jbatx = 0.0678}$$

Step 3. Derive Limit Qual-Test Rigid-Body Accelerations

Design criterion: Assuming all critical failure modes are ductile, we will design to a load of 3 times the rms level.

A. Box:

Axial load: RMS acceleration (g), $Azrms := 35$

Limit load (g), $Azboxq := 3 \cdot Azrms$ $\boxed{Azboxq = 105}$

Lateral load:

Z distance of measured acceleration on top panel (in.), $zbox = 12.5$

Z distance of box c.g. to critical interface (in), $Zbarbox = 6.25$ (from above)

Measured rms acceleration (g) at top of box at qual level, $Axtoprms := 19.8$

(ref. Table E-1, Appendix E)

Note: Axtoprms is the rms acceleration associated only with the 154-Hz fundamental lateral mode, equal to the square root of the area under that part of the response PSD.

Assuming zero acceleration at FS2/ring interface, the limit c.g. acceleration is

$$Axboxq := 3 \cdot Axtoprms \cdot \frac{Zbarbox}{zbox} \qquad \boxed{Axboxq = 29.7}$$

To calculate the rigid-body rotational acceleration, first convert the measured top acceleration to units of inches per second squared:

$$g := 386.1 \qquad Atop1 := Axtoprms \cdot g \qquad Atop1 = 7.6448 \times 10^{3}$$

Based on small-angle theory, the limit rotational acceleration (rad/s2) is

$$Ryboxq := 3 \cdot \frac{Atop1}{zbox} \qquad \boxed{Ryboxq = 1.8347 \times 10^{3}}$$

Step 3. Derive Limit Qual-Test Rigid-Body Accelerations (continued)

B. Column:

Axial load: insignificant
Lateral load:

Z distance of measured acceleration (in.), $\mathrm{zcol} = 8$

Z distance of c.g. to critical interface (in), $\mathrm{Zbarcol} = 4$ (from above)

Measured rms acceleration (g) at top of column at qual level, $\mathrm{Axcolrms} := 20.1$

(ref. Sec. E.2d)

Assuming zero acceleration at column/baseplate interface, the limit c.g. acceleration is

$$\mathrm{Axcolq} := 3 \cdot \mathrm{Axcolrms} \cdot \frac{\mathrm{Zbarcol}}{\mathrm{zcol}} \qquad \boxed{\mathrm{Axcolq} = 30.15}$$

measured acceleration to units of inches per second squared:

$$g := 386.1 \qquad \mathrm{Acolg} := \mathrm{Axcolrms} \cdot g \qquad \mathrm{Acolg} = 7.7606 \times 10^{3}$$

Based on small-angle theory, the limit rotational acceleration (rad/s2) is

$$\mathrm{Rycolq} := 3 \cdot \frac{\mathrm{Acolg}}{\mathrm{zcol}} \qquad \boxed{\mathrm{Rycolq} = 2.9102 \times 10^{3}}$$

Step 3. Derive Limit Qual-Test Rigid-Body Accelerations (continued)

C. Battery Pack

Axial load: insignificant

Lateral load--X direction:

Z distance of measured acceleration (in.), $zbat = 7.05$

Z distance of c.g. to critical interface (in), $Zbarbat = 3.525$ (from above)

Measured rms acceleration (g) at top of battery at qual level, $Axbatrms := 44$

(ref. Sec. E.2e)

Assuming zero acceleration at battery/baseplate interface, the limit c.g. acceleration is

$$Axbatq := 3 \cdot Axbatrms \cdot \frac{Zbarbat}{zbat} \qquad \boxed{Axbatq = 66}$$

measured acceleration to units of inches per second squared:

$g := 386.1$ $\quad Abatg := Axbatrms \cdot g$ $\quad Abatg = 1.6988 \times 10^{4}$

Based on small-angle theory, the limit rotational acceleration (rad/s2) is

$$Rybatq := 3 \cdot \frac{Abatg}{zbat} \qquad \boxed{Rybatq = 7.2291 \times 10^{3}}$$

Lateral load--Y direction:

Measured rms acceleration (g) at top of battery at qual level, $Aybatrms := 62$

(ref. Sec. E.2f)

Assuming zero acceleration at battery/baseplate interface, the limit c.g. acceleration is

$$Aybatq := 3 \cdot Aybatrms \cdot \frac{Zbarbat}{zbat} \qquad \boxed{Aybatq = 93}$$

measured acceleration to units of inches per second squared:

$g := 386.1$ $\quad Abatg := Aybatrms \cdot g$ $\quad Abatg = 2.3938 \times 10^{4}$

Based on small-angle theory, the limit rotational acceleration (rad/s2) is

$$Rxbatq := 3 \cdot \frac{Abatg}{zbat} \qquad \boxed{Rxbatq = 1.0186 \times 10^{4}}$$

Step 4. Calculate Limit Forces and Moments

A. FS2/ring interface

Limit axial force (lb):

$fzringq := Azboxq \cdot Wbox$ $\quad$ $\boxed{fzringq = 2.4707 \times 10^3}$ (qual)

$fzring := \dfrac{fzringq}{1.4}$ $\quad$ $\boxed{fzring = 1.7648 \times 10^3}$ (flight)

Limit shear force (lb):

$vringq := Axboxq \cdot Wbox$ $\quad$ $\boxed{vringq = 698.841}$ (qual)

$vring := \dfrac{vringq}{1.4}$ $\quad$ $\boxed{vring = 499.1721}$ (flight)

Limit moment (in-lb):

$mringq := Axboxq \cdot Wbox \cdot Zbarbox + Ryboxq \cdot Jboxx$ $\quad$ $\boxed{mringq = 8.685 \times 10^3}$ (qual)

$mring := \dfrac{mringq}{1.4}$ $\quad$ $\boxed{mring = 6.2036 \times 10^3}$ (flight)

Compare the above flight limit loads to the shear and moment predicted for 25-g limit uniform acceleration, which envelops the specified quasi-static loads (see Appendix A of the SVP):

For 25-g limit uniform lateral load,

$fzringL := 25 \cdot (Wbox + Wcol + Wbat)$ $\quad$ $\boxed{fzringL = 957.75}$

$vringL := 25 \cdot (Wbox + Wcol + Wbat)$ $\quad$ $\boxed{vringL = 957.75}$

$mringL := 25 \cdot (Wbox \cdot Zbarbox + Wcol \cdot Zbarcol + Wbat \cdot Zbarbat)$ $\quad$ $\boxed{mringL = 5.1033 \times 10^3}$

Conclusion: Peak axial force and moment come from random vibration, whereas peak shear comes from the 25-g quasi-static load.

Step 4. Calculate Limit Forces and Moments (continued)

B. Column/baseplate interface

Limit shear force (lbs):

$vcolq := Axcolq \cdot Wcol$ $\quad \boxed{vcolq = 315.369}$ (qual)

$vcol := \dfrac{vcolq}{1.4}$ $\quad \boxed{vcol = 225.2636}$ (flight)

Limit moment (in-lb):

$mcolq := Axcolq \cdot Wcol + Rycolq \cdot Jcolx$ $\quad \boxed{mcolq = 1.4044 \times 10^{3}}$ (qual)

$mcol := \dfrac{mcolq}{1.4}$ $\quad \boxed{mcol = 1.0031 \times 10^{3}}$ (flight)

C. Battery/baseplate interface

X direction:

Limit shear force (lbs):

$vxbatq := Axbatq \cdot Wbat$ $\quad \boxed{vxbatq = 285.12}$ (qual)

$vxbat := \dfrac{vxbatq}{1.4}$ $\quad \boxed{vxbat = 203.6571}$ (flight)

Limit moment (in-lb):

$mybatq := Axbatq \cdot Wbat + Rybatq \cdot Jbatx$ $\quad \boxed{mybatq = 775.4354}$ (qual)

$mybat := \dfrac{mybatq}{1.4}$ $\quad \boxed{mybat = 553.8824}$ (flight)

Y direction:

Limit shear force (lbs):

$vybatq := Aybatq \cdot Wbat$ $\quad \boxed{vybatq = 401.76}$ (qual)

$vybat := \dfrac{vybatq}{1.4}$ $\quad \boxed{vybat = 286.9714}$ (flight)

Limit moment (in-lb):

$mxbatq := Aybatq \cdot Wbat + Rxbatq \cdot Jbatx$ $\quad \boxed{mxbatq = 1.0927 \times 10^{3}}$ (qual)

$mxbat := \dfrac{mxbatq}{1.4}$ $\quad \boxed{mxbat = 780.4707}$ (flight)

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Appendix B
Strength Analysis Details

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May 13, 2002

B.1 Introduction:

The baseplate is the critical structural part for FalconSat-2 (FS2) because all structural loads funnel through it into the mounting structure (adapter ring). The following analysis assesses the baseplate's strength under the design loads derived in Appendix A. The bolt patterns shown below are also assessed. Margins of safety (MS) are calculated for flight loads and for the qualification random-vibration loads. The other structural parts in FS2--the side panels (other than bearing under bolt shear at the base-plate interface), the top panel (except for ground-handling loads), and the column--are okay by inspection for flight loads. Stresses are low in these parts. At the end of this appendix is an assessment of the top

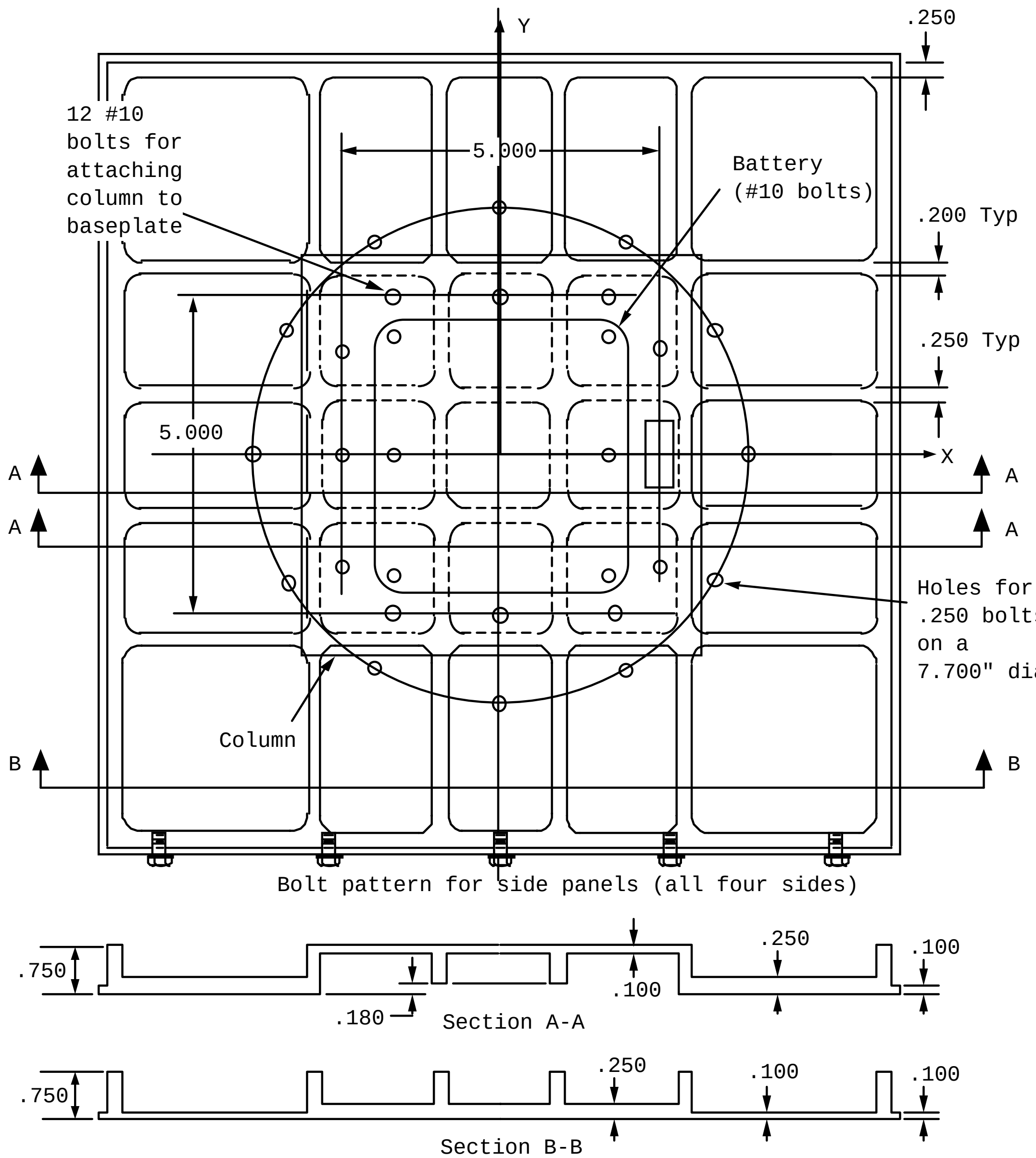


Fig. B-1. Base-Plate Geometry.

B.2 Potential Failure Modes and Margin-of-Safety Summary

		Qual MS	Flight MS	Ref Page
Base Plate:				
1. Bolt circle attaching baseplate to adapter ring				
1a. Bolt/nut tension, ultimate		Large*	Large*	B-7
1b. Nut failure, ultimate		Large*	Large*	B-7
1c. End-pad shear, ultimate		Large*	Large*	B-7
1d. End-pad bending, ultimate		3.62	4.84	B-8
1d. End-pad bending, yield		2.88	3.99	B-8
1e. Bolt shear and bearing, ultimate		Large*	Large*	B-6
2. Bolts attaching column to baseplate				
2a. Bolt tension, ultimate		Large*	Large*	B-10
2b. Failure of insert, ultimate		Large*	Large*	B-10
2c. End-pad shear, ultimate		Large*	Large*	B-10
2d. End-pad bending, ultimate		4.30	Large*	B-11
2d. End-pad bending, yield		3.44	4.71	B-11
2e. Bolt shear and bearing, ultimate		Large*	Large*	B-10
3. Bolts attaching battery to baseplate				
3a. Bolt tension, ultimate		Large*	Large*	B-12
3b. Nut failure, ultimate		Large*	Large*	B-12
3c. End-pad shear, ultimate		Large*	Large*	B-12
3d. End-pad bending, ultimate		0.52	0.92	B-14
3d. End-pad bending, yield		0.28	0.64	B-14
3e. Bolt shear and bearing, ultimate		Large*	Large*	B-12
4. Bending of machined ribs	Ult:	1.57	2.24	B-19
	Yld:	0.87	1.41	B-19
5. Bolts attaching side walls to baseplate				
5a. Bolt shear, ult		3.21	4.32	B-20
5b. Bearing, ult		1.79	2.53	B-20
yld		2.00	2.85	B-20

Top Panel for Ground-handling Loads:	MS	Page
1. End-pad bending, ultimate	0.59	B-22
yield	0.39	B-23
2. Rib bending, ultimate	1.50	B-23
yield	1.19	B-23

*"Large" margins of safety are above 500%

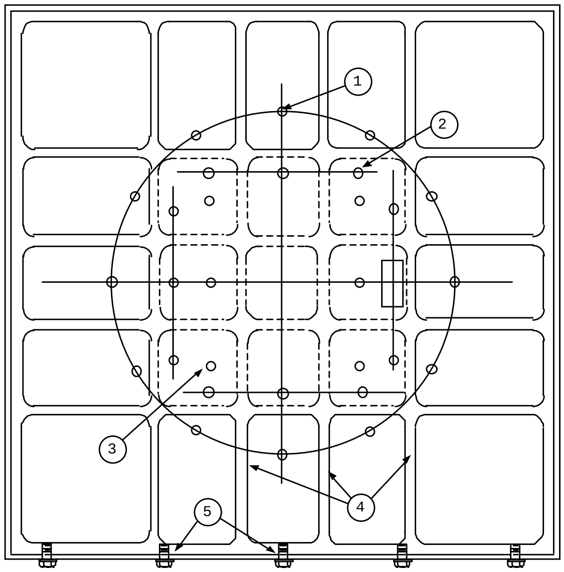


Fig. B-2. Failure Mode Locations for Base-Plate.

Baseplate material:
6061-T651 Plate

Bolts for failure mode #1:
NAS1351, 1/4" dia., 160 ksi
Nuts: NAS1805, 160 ksi

Bolts for failure modes 2 and 5:
NAS1351, .190" dia., 160 ksi
Inserts: 300-series SS Spiralock

Bolts for failure mode 3:
NAS1351, .190" dia., 160 ksi
Nuts: NAS1805, 160 ksi

B.3 Reference Data

Tensile-stress area (in2), *At*, for UN bolts:

Bolt Diameter (in.)	At (fine threads)
0.190	0.0200
0.250	0.0364

Material Properties (stresses in units of psi):

$Ftub := 160000$ (allowable tensile ultimate stress for bolt material)

$Fsub := 95000$ (allowable shear ultimate stress for bolt material)

$Ftuf := 42000$ (allowable tensile ultimate stress for baseplate)

$Fbuf := 60000$ (allowable plastic bending ultimate stress for baseplate)

$Fsuf := 27000$ (allowable shear ultimate stress for baseplate)

$Fcy := 35000$ (allowable compressive yield stress for baseplate)

$Fbru := 67000$ (allowable bearing ultimate stress for baseplate, e/d = 1.5)

$Fbry := 50000$ (allowable bearing yield stress for baseplate, e/d = 1.5)

$e := 9$ (elongation of baseplate material, %, in the LT direction)

Materials properties are from MIL-HDBK-5G. Allowable stresses are A-basis.

Criteria:

Factors of safety:	For qual loads:	$FSqu := 1.25$ (ult)	$FSqy := 1.0$ (yld)
	For flight loads:	$FSfu := 1.4$ (ult)	$FSfy := 1.1$ (yld)

Fiting factor: $FF := 1.15$

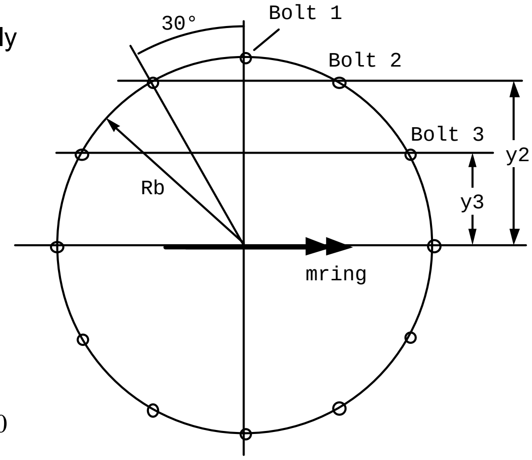
B.4 Analysis of Base Plate

Failure Mode 1: Bolt circle attaching base plate to ring

First step: Calculate bolt loads from applied loads defined at centroid of bolt pattern:

Given: 12 bolts, 1/4 dia, evenly spaced at 30 degrees

Radius of bolt circle (in), $Rb := 3.85$



Limit forces (lb) and moments (in.lb) at centroid of bolt pattern for qualification testing (from Appendx A):

Shear force (lbs), $vringq := 699$

Overturning moment (in-lb), $mringq := 8690$

$y2 := Rb \cdot \sin(60 \cdot deg)$

$y3 := Rb \cdot \sin(30 \cdot deg)$

Calculate tension in bolts 1 (pt1), 2 (pt2), and 3 (pt3) in pounds, assuming bolt loads caused by interface moment are proportional to distance from neutral axis:

For moment m, pt1 = a*mringq, where $a := \dfrac{1}{2 \cdot Rb + 4 \cdot \dfrac{y2^2}{Rb} + 4 \cdot \dfrac{y3^2}{Rb}}$ $\quad a = 0.043$

pt2 = b*mringq, where $b := \dfrac{y2}{Rb} \cdot a$ $\quad b = 0.037$

pt3 = c*mringq, where $c := \dfrac{y3}{Rb} \cdot a$ $\quad c = 0.022$

For the full set of specified limit loads, the limit bolt tensile loads (lbs) are

$pt1 := a \cdot mringq$ $\quad \boxed{pt1 = 376.19}$

$pt2 := b \cdot mringq$ $\quad \boxed{pt2 = 325.791}$

$pt3 := c \cdot mringq$ $\quad \boxed{pt3 = 188.095}$

Limit bolt shear force (lbs):

Because FS2 is attached to a cylindrical adapter, bolt shear loads should vary according to the way in which shear flows through a cylinder, with loads acting tangentially. Thus, assume the peak bolt shear is twice the value calculated by dividing the applied shear by 12 bolts.

$ps4 := 2 \cdot \dfrac{vringq}{12}$ $\quad \boxed{ps4 = 116.5}$

Failure Mode 1: Bolt circle attaching base plate to ring (continued)

Design Bolt Loads:

Qual test:

Design ultimate bolt tensile load, lb: $ptuq := FSqu \cdot FF \cdot pt1$ $\boxed{ptuq = 540.774}$

Design yield bolt tensile load, lb: $ptyq := FSqy \cdot pt1$ $\boxed{ptyq = 376.19}$

Design ultimate bolt shear load, lb: $psuq := FSqu \cdot FF \cdot ps4$ $\boxed{psuq = 167.469}$

Flight:

Design ultimate bolt tensile load, lb: $ptuf := 0.707 \cdot pt1 \cdot FSfu \cdot FF$ $\boxed{ptuf = 428.206}$

Design yield bolt tensile load for flight, lb: $ptyf := FSfy \cdot 0.707 \cdot pt1$ $\boxed{ptyf = 292.563}$

Design ultimate bolt shear load, lb: $psuf := 0.707 \cdot ps4 \cdot FSfu \cdot FF$ $\boxed{psuf = 132.608}$

Bolt shear and bearing are okay by inspection; MS = large

Analysis of a Tension Joint with a Single Bolt: Channel-Type Fitting

1a. Bolt tension (ultimate only)
1b. Nut failure (ultimate only)
1c. End-pad shear (ultimate only)
1d. End-pad bending (ultimate and yield, equations derived from Lockheed Stress Memo #88a)

Prerequisite assumptions and limitations (which are met):

- The fitting material is ductile (to be able to rely on the end-pad bending method)
- The bolt material is ductile (to ensure preload does not contribute to tensile failure)

Inputs:

$Db := .250$ (bolt diameter)

$At := .0364$ (tensile-stress area from above table)

$Dbh := .450$ (diameter of bolt head)

$a := 1.25$ $\quad Tb := .20$

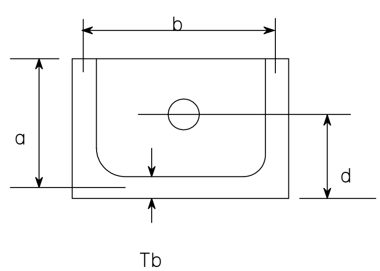
$b := 1.90$ $\quad Te := .25$

$d := 1.10$

Calculated:

$ri := \frac{Db}{2}$ $\quad ri = 0.125$

(bolt-hole radius)



Note: minimum thickness is 0.24", and 1.05 times 0.24 is greater than 0.25, so the

Failure Mode 1: Bolt circle attaching base plate to ring (continued)

Analysis Process:

For each failure mode, calculate an allowable bolt load, which is the bolt load that would cause the allowable stress for that failure mode.

Compute margins of safety by comparing the calculated allowable bolt loads to the design bolt loads.

Calculation of Allowables and Margins of Safety

1a. Bolt tensile failure

Allowable ult. bolt tensile load, $Ptu1a := Ftub \cdot At$ $\qquad Ptu1a = 5.824 \times 10^{3}$

$$MSuq1a := \frac{Ptu1a}{ptuq} - 1 \qquad \boxed{MSuq1a = 9.77} \text{ (qual)}$$

$$MSuf1a := \frac{Ptu1a}{ptuf} - 1 \qquad \boxed{MSuf1a = 12.601} \text{ (flight)}$$

1b. Nut failure

The selected nut (NAS1805) is a 160-ksi nut. It can fully develop the strength of the bolt, so its failure is of no concern.

1c. End-pad shear failure

For an channel-type fitting like this, assume the shear area, As, equals the end-pad thickness times 75% of the circumference of the bolt head:

$As := Te \cdot .75 \cdot \pi \cdot Dbh \qquad As = 0.265$

Allowable ult. bolt tensile load for end-pad shear, $Ptu1c := Fsuf \cdot As \qquad Ptu1c = 7.157 \times 10^{3}$

$$MSuq1c := \frac{Ptu1c}{ptuq} - 1 \qquad \boxed{MSuq1c = 12.235} \text{ (qual)}$$

$$MSuf1c := \frac{Ptu1c}{ptuf} - 1 \qquad \boxed{MSuf1c = 15.714} \text{ (flight)}$$

Failure Mode 1: Bolt circle attaching base plate to ring (continued)

1d. End-pad bending failure

$$\frac{ri}{a} = 0.1$$ (Limitation: 0.1 < ri/a < 0.4)

$$\frac{b}{a} = 1.52$$ (Limitation: between 1.0 and 3.0)

$$K3 := -3.81 + \frac{3.8}{\left(\frac{ri}{a} + 0.015\right)^{.1}} + 0.15 \cdot \ln\left(\frac{b}{a} - 0.30\right)$$

Note: The equation for K3 was developed in 1983 by Tom Sarafin to fit the curve used in Lockheed Stress Memo 88a. Staying within the above limitations, the calculated bending stress agrees with that calculated using the Stress Memo to within the accuracy possible by manually using the Memo curves.

Allowable ult. bolt tensile load for end-pad bending,

$$Ptu1d := \frac{\left(Fbuf \cdot a \cdot Te^2\right)}{K3 \cdot (2d - Tb)}$$

$$Ptu1d = 2.5 \times 10^3$$

$$MSuq1d := \frac{Ptu1d}{ptuq} - 1$$ $\boxed{MSuq1d = 3.624}$ (qual)

$$MSuf1d := \frac{Ptu1d}{ptuf} - 1$$ $\boxed{MSuf1d = 4.839}$ (flight"

Allowable yield bolt tensile load for end-pad bending,

$$Pty1d := \frac{\left(Fcy \cdot a \cdot Te^2\right)}{K3 \cdot (2d - Tb)} \qquad Pty1d = 1.459 \times 10^3$$

$$MSyq1d := \frac{Pty1d}{ptyq} - 1$$ $\boxed{MSyq1d = 2.877}$ (qual)

$$MSyf1d := \frac{Pty1d}{ptyf} - 1$$ $\boxed{MSyf1d = 3.985}$ (flight)

1e. Bolt shear and bearing:

Bolt shear and bearing okay by inspection; MS = large

Failure Mode 2: Bolts attaching column to baseplate

First step: Calculate bolt loads from applied loads defined at centroid of bolt pattern:

Given:

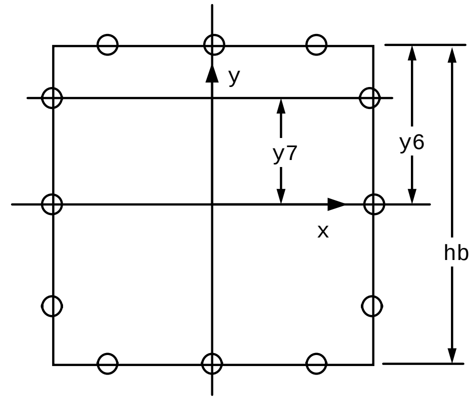
$hb := 5.00$

$y6 := \frac{hb}{2}$

$y7 := 1.50$

12 bolts, 160 ksi, #10 (NAS1351)

Spiralock inserts, one-diameter length (1900 lb allowable ult. pull-out strength, ref. Appendix F)



Limit forces (lb) and moments (in.lb) at centroid of bolt pattern for qualification testing (from Apdx A):

Shear force (lbs), $vcolq := 315$

Overturning moment (in-lb), $mcolq := 1400$

Calculate tension (lb) in the bolts farthest from the neutral axis (pt6) and in the bolts that are a distance of y7 from the neutral axis (pt7), assuming bolt loads caused by interface moment are proportional to distance from neutral axis:

For moment mcol, pt6 = a*mcol, where $a := \frac{1}{6 \cdot y6 + 4 \cdot \frac{y7^2}{y6}}$ $a = 0.054$

pt7 = b*mcol, where $b := \frac{y7}{y6} \cdot a$ $b = 0.032$

For the full set of specified limit loads, the limit bolt tensile loads (lbs) are

$pt6 := a \cdot mcolq$ $\boxed{pt6 = 75.269}$

$pt7 := b \cdot mcolq$ $\boxed{pt7 = 45.161}$

Limit bolt shear force (lbs):

Because the column has a thin-walled square cross section, in which shear is carried only by the two walls parallel to the applied shear, assume only half the bolts carry the shear:

$ps7 := \frac{vcolq}{6}$ $\boxed{ps7 = 52.5}$

Failure Mode 2: Bolts attaching column to baseplate (continued)

Criteria:

Factors of safety: For qual loads: $FSqu := 1.25$ (ult) $FSqy := 1.0$ (yld)

For flight loads: $FSfu := 1.4$ (ult) $FSfy := 1.1$ (yld)

Fiting factor: $FF := 1.15$ (applicable for ultimate only, because alignment is not critical)

Design Bolt Loads:

Qual test:

Design ultimate bolt tensile load, lb: $ptuq := FSqu \cdot FF \cdot pt6$ $\boxed{ptuq = 108.199}$

Design yield bolt tensile load, lb: $ptyq := FSqy \cdot pt6$ $\boxed{ptyq = 75.269}$

Design ultimate bolt shear load, lb: $psuq := FSqu \cdot FF \cdot ps7$ $\boxed{psuq = 75.469}$

Flight:

Design ultimate bolt tensile load, lb: $ptuf := 0.707 \cdot pt6 \cdot FSfu \cdot FF$ $\boxed{ptuf = 85.676}$

Design yield bolt tensile load for flight, lb: $ptyf := FSfy \cdot 0.707 \cdot pt6$ $\boxed{ptyf = 58.537}$

Design ultimate bolt shear load, lb: $psuf := 0.707 \cdot ps7 \cdot FSfu \cdot FF$ $\boxed{psuf = 59.759}$

Bolt tension, insert failure, end-pad shear, and bolt shear are okay by inspection; MS = large

Analysis of a Tension Joint with a Single Bolt: Channel-Type Fitting

2d. End-pad bending (ultimate and yield, equations derived from Lockheed Stress Memo #88a)

Prerequisite assumption and limitation (which is met):

- The fitting material is ductile (to be able to rely on the end-pad bending method)

Inputs:

$Db := .190$ (bolt diameter)

$At := .0200$ (tensile-stress area from above table)

$Dbh := .340$ (diameter of bolt head)

$a := 0.95$ $Tb := .20$

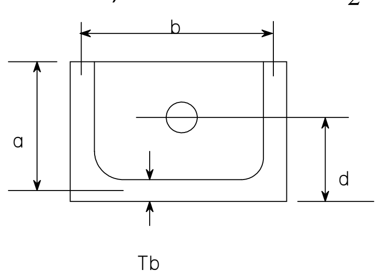
$b := 1.90$ $Te := 1.05 \cdot (.100 - 0.01)$

$d := 0.55$ $Te = 0.095$

Calculated:

$ri := \frac{Db}{2}$ $ri = 0.095$

(bolt-hole radius)



Failure Mode 2: Bolts attaching column to baseplate (continued)

Analysis Process:

For the end-pad bending failure mode, calculate allowable bolt loads, which are the bolt loads that would cause the allowable stresses for that failure mode.

Compute margins of safety by comparing the calculated allowable bolt loads to the design bolt loads.

Calculation of Allowables and Margins of Safety

2d. End-pad bending failure

$$\frac{ri}{a} = 0.1$$ (Limitation: 0.1 < ri/a < 0.4)

$$\frac{b}{a} = 2$$ (Limitation: between 1.0 and 3.0)

$$K3 := -3.81 + \frac{3.8}{\left(\frac{ri}{a} + 0.015\right)^{.1}} + 0.15 \cdot \ln\left(\frac{b}{a} - 0.30\right)$$

Note: The equation for K3 was developed in 1983 by Tom Sarafin to fit the curve used in Lockheed Stress Memo 88a. Staying within the above limitations, the calculated bending stress agrees with that calculated using the Stress Memo to within the accuracy possible by manually using the Memo curves.

Allowable ult. bolt tensile load for end-pad bending,

$$Ptu2d := \frac{\left(Fbuf \cdot a \cdot Te^2\right)}{K3 \cdot (2d - Tb)}$$

$$Ptu2d = 572.965$$

$$MSuq2d := \frac{Ptu2d}{ptuq} - 1 \qquad \boxed{MSuq2d = 4.295}$$ (qual)

$$MSuf2d := \frac{Ptu2d}{ptuf} - 1 \qquad \boxed{MSuf2d = 5.688}$$ (flight"

Allowable yield bolt tensile load for end-pad bending,

$$Pty2d := \frac{\left(Fcy \cdot a \cdot Te^2\right)}{K3 \cdot (2d - Tb)} \qquad Pty2d = 334.23$$

$$MSyq2d := \frac{Pty2d}{ptyq} - 1 \qquad \boxed{MSyq2d = 3.44}$$ (qual)

$$MSyf2d := \frac{Pty2d}{ptyf} - 1 \qquad \boxed{MSyf2d = 4.71}$$ (flight)

Failure Mode 3: Bolts attaching battery pack to baseplate

First step: Calculate bolt loads from applied loads defined at centroid of bolt pattern:

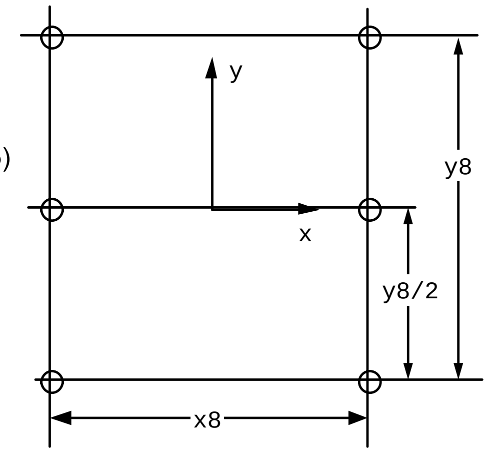
Given:

6 bolts, 160 ksi, #10 (NAS1351)

$y8 := 3.70$

160-ksi nuts (NAS1805)

$x8 := 3.39$



Limit forces (lb) and moments (in.lb) at centroid of bolt pattern for qualification testing (from Apdx A):

Shear force (lbs), $vybatq := 402$

Overturning moment (in-lb), $mxbatq := 1090$

Calculate limit tension (lb) in the bolts farthest from the neutral axis (pt8):

$$pt8 := \frac{mxbatq}{2 \cdot y8} \qquad \boxed{pt8 = 147.297}$$

Limit bolt shear force (lbs):

$$ps8 := \frac{vybatq}{6} \qquad \boxed{ps8 = 67}$$

Criteria:

Factors of safety: For qual loads: $FSqu := 1.25$ (ult) $FSqy := 1.0$ (yld)

For flight loads: $FSfu := 1.4$ (ult) $FSfy := 1.1$ (yld)

Fiting factor: $FF := 1.15$ (applicable for ultimate only, because alignment is not critical)

Design Bolt Loads:

Qual test:

Design ultimate bolt tensile load, lb: $ptuq := FSqu \cdot FF \cdot pt8$ $\boxed{ptuq = 211.74}$

Design yield bolt tensile load, lb: $ptyq := FSqy \cdot pt8$ $\boxed{ptyq = 147.297}$

Design ultimate bolt shear load, lb: $psuq := FSqu \cdot FF \cdot ps8$ $\boxed{psuq = 96.313}$

Flight:

Design ultimate bolt tensile load, lb: $ptuf := 0.707 \cdot pt8 \cdot FSfu \cdot FF$ $\boxed{ptuf = 167.664}$

Design yield bolt tensile load for flight, lb: $ptyf := FSfy \cdot 0.707 \cdot pt8$ $\boxed{ptyf = 114.553}$

Design ultimate bolt shear load, lb: $psuf := 0.707 \cdot ps8 \cdot FSfu \cdot FF$ $\boxed{psuf = 76.264}$

Bolt tension, nut failure, end-pad shear, and bolt shear are okay by inspection; MS = large

Failure Mode 3: Bolts attaching battery pack to baseplate (continued)

Analysis of a Tension Joint with a Single Bolt: Channel-Type Fitting

3d. End-pad bending (ultimate and yield, equations derived from Lockheed Stress Memo #88a)

Prerequisite assumption and limitation (which is met):

- The fitting material is ductile (to be able to rely on the end-pad bending method)

Inputs:

$Db := .190$ (bolt diameter)

$At := .0200$ (tensile-stress area from above table)

$Dbh := .340$ (diameter of bolt head)

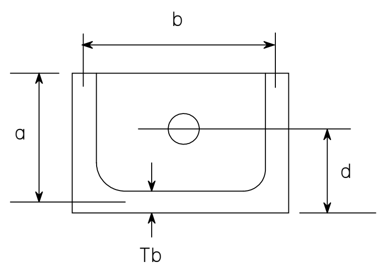
$a := 0.95$ $\quad Tb := .20$

$b := 1.90$

$d := 0.90$

$Te := 1.05\cdot(.100 - 0.0$

$Te = 0.095$



Calculated:

$ri := \frac{Db}{2}$ $\quad ri = 0.095$

(bolt-hole radius)

Analysis Process:

For the end-pad bending failure mode, calculate allowable bolt loads, which are the bolt loads that would cause the allowable stresses for that failure mode.

Compute margins of safety by comparing the calculated allowable bolt loads to the design bolt loads.

Failure Mode 3: Bolts attaching battery pack to baseplate (continued)

Calculation of Allowables and Margins of Safety

3d. End-pad bending failure

$$\frac{ri}{a} = 0.1$$ (Limitation: 0.1 < ri/a < 0.4)

$$\frac{b}{a} = 2$$ (Limitation: between 1.0 and 3.0)

$$K3 := -3.81 + \frac{3.8}{\left(\frac{ri}{a} + 0.015\right)^{.1}} + 0.15 \cdot \ln\left(\frac{b}{a} - 0.30\right)$$

Note: The equation for K3 was developed in 1983 by Tom Sarafin to fit the curve used in Lockheed Stress Memo 88a. Staying within the above limitations, the calculated bending stress agrees with that calculated using the Stress Memo to within the accuracy possible by manually using the Memo curves.

Allowable ult. bolt tensile load for end-pad bending,

$$Ptu3d := \frac{\left(Fbuf \cdot a \cdot Te^2\right)}{K3 \cdot (2d - Tb)}$$

$$Ptu3d = 322.293$$

$$MSuq3d := \frac{Ptu3d}{ptuq} - 1 \qquad \boxed{MSuq3d = 0.522}$$ (qual)

$$MSuf3d := \frac{Ptu3d}{ptuf} - 1 \qquad \boxed{MSuf3d = 0.922}$$ (flight"

Allowable yield bolt tensile load for end-pad bending,

$$Pty3d := \frac{\left(Fcy \cdot a \cdot Te^2\right)}{K3 \cdot (2d - Tb)} \qquad Pty3d = 188.004$$

$$MSyq3d := \frac{Pty3d}{ptyq} - 1 \qquad \boxed{MSyq3d = 0.276}$$ (qual)

$$MSyf3d := \frac{Pty3d}{ptyf} - 1 \qquad \boxed{MSyf3d = 0.641}$$ (flight)

Failure Mode 4: Bending of Baseplate Ribs

When the fundamental rocking mode for FS-2 is excited, with rotation about the X axis, , the box rotates and causes a bending moment, mring, at the FS-2/ring interface. The ribs in the baseplate bend in this situation, transfering shear from the box walls to the bolt circle. A section of width b1, below, is assumed effective in carrying this bending moment.

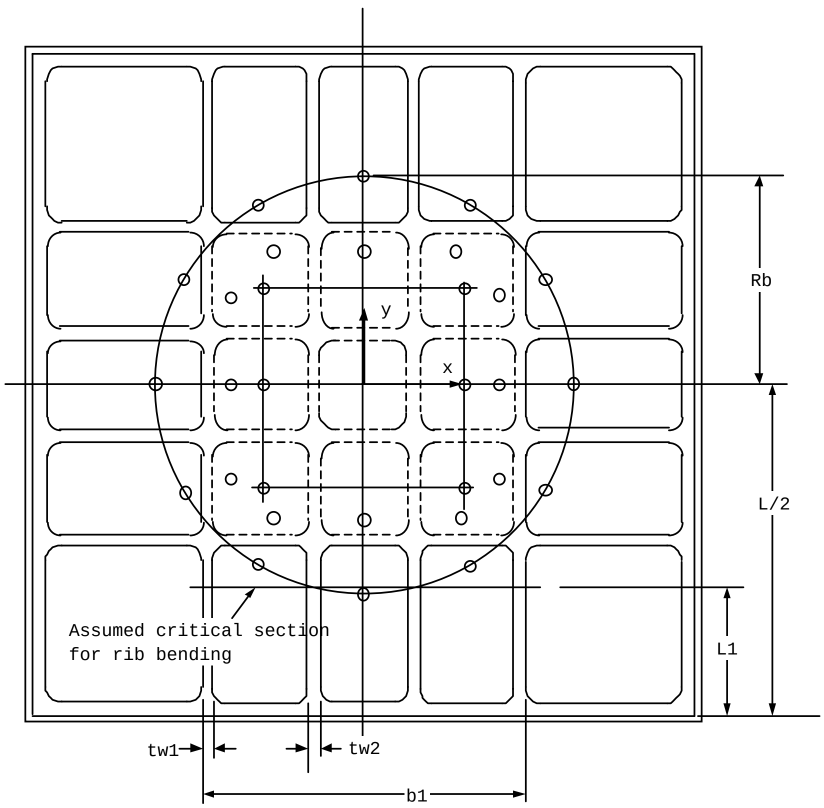
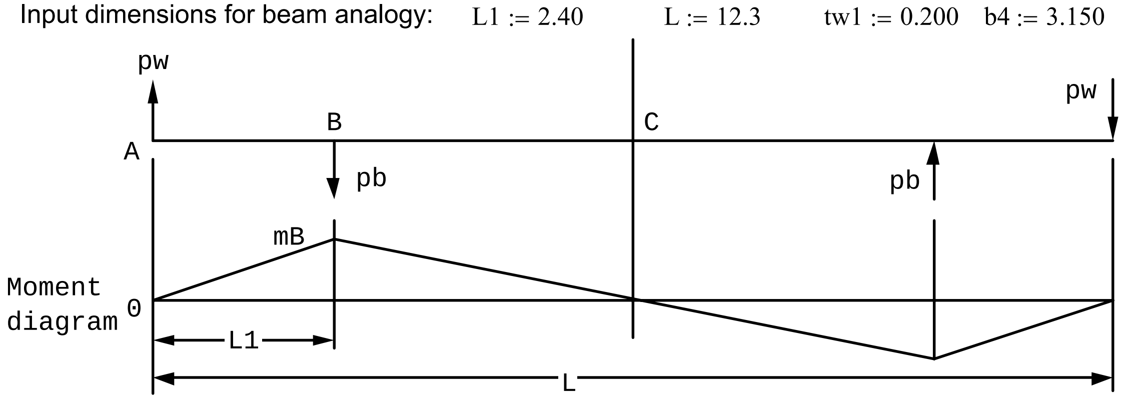


Fig. B-3. Base Plate Dimensions for Rib-Bending Check.

Input dimensions for beam analogy: $L1 := 2.40$ $L := 12.3$ $tw1 := 0.200$ $b4 := 3.150$



The bending moment peaks at point B.

Failure Mode 4: Bending of Baseplate Ribs (continued)

Step 1: Calculate peak bending moment, mB

The loads pw shown on the beam diagram are forces introduced by the wall of the box caused by the maximum response of the fundamental rocking mode. Assume the couple formed by these loads is equal to the moment introduced by the inertia of the box. From the above assessment of failure mode 1, the limit moment (in-lb) for qual test is

$$mringq = 8.69 \times 10^3$$

Thus,

$$pw := \frac{mringq}{L} \qquad pw = 706.504 \quad \text{(Limit, qual test)}$$

$$pb := \frac{mringq}{L - 2 \cdot L1} \qquad pb = 1.159 \times 10^3$$

And

$$mB := pw \cdot L1 \qquad \boxed{mB = 1.696 \times 10^3} \quad \text{(Limit, qual test)}$$

Step 2: Calculate section properties for assumed section (Fig. B-4)

$b1 := 6.00$ $\qquad tw1 := 0.20$

$a2 := 0.50$ $\qquad tw2 := 0.25$

$bh := 0.50$ $\qquad t3 := 0.25$

(Analysis based on nominal thickness)

The section shown is assumed effective in bending. Material of width bh has been removed at each bolt hole in this analysis, based on the assumption that this material is highly stressed from local end-pad bending and is thus not effective for beam bending.

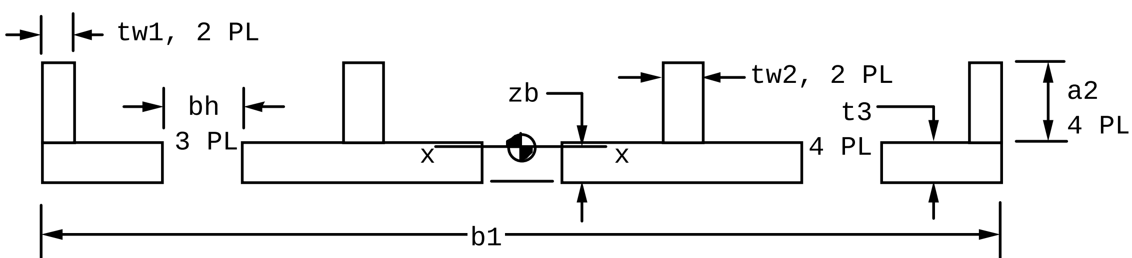


Fig. B-4. Bending Section.

Area, A (in^2):

$$A1 := 2 \cdot a2 \cdot tw1$$

$$A2 := 2 \cdot a2 \cdot tw2 \qquad bnet := b1 - 3 \cdot bh$$

$$A3 := (bnet) \cdot t3 \qquad A := A1 + A2 + A3 \qquad \boxed{A = 1.575}$$

Centroid location, zb (in):

$$zb := \frac{1}{A} \cdot \left[A3 \cdot \frac{t3}{2} + A2 \cdot \left(t3 + \frac{a2}{2} \right) + A1 \cdot \left(t3 + \frac{a2}{2} \right) \right] \qquad zb = 0.232$$

Failure Mode 4: Bending of Baseplate Ribs (continued)

Centroidal moment of inertia, Ix (in^4):

$$Ix1 := A3\cdot\left(\frac{t3}{2} - zb\right)^2 + A2\cdot\left(t3 + \frac{a2}{2} - zb\right)^2 + A1\cdot\left(t3 + \frac{a2}{2} - zb\right)^2$$

$$Ix2 := \frac{1}{12}\cdot\left[(bnet)\cdot t3^3 + 2\cdot tw2\cdot a2^3 + 2\cdot tw1\cdot a2^3\right]$$

$$Ix := Ix1 + Ix2 \qquad \boxed{Ix = 0.06}$$

Calculate the limit bending stress (psi) in the rib section for qualification random vibration:

$$fbq := mB\cdot\frac{(a2 + t3 - zb)}{Ix} \qquad fbq = 1.453\times 10^4$$

For flight:

$$fbf := .707\cdot fbq \qquad fbf = 1.027\times 10^4$$

Allowable bending stresses:

Yield: $Fcy = 3.5\times 10^4$

Ultimate: $Fbuf = 6\times 10^4$

Margins of safety for qual testing:

$$MSyq4 := \frac{Fcy}{FSqy\cdot fbq} - 1 \qquad \boxed{MSyq4 = 1.409}$$

$$MSuq4 := \frac{Fbuf}{FSqu\cdot fbq} - 1 \qquad \boxed{MSuq4 = 2.304}$$

Margins of safety for flight:

$$MSyf4 := \frac{Fcy}{FSfy\cdot fbf} - 1 \qquad \boxed{MSyf4 = 2.097}$$

$$MSuf4 := \frac{Fbuf}{FSfu\cdot fbf} - 1 \qquad \boxed{MSuf4 = 3.172}$$

Failure Mode 4: Bending of Baseplate Ribs (continued)

Now check the ribs in the interior of the base plate, where the plate thickness is reduced by 0.180" to ensure the separation switches will mate properly with NASA hardware:

The reduced-thickness section starts at a distance of L5 from the edge of the plate:

$$L5 := b4 + tw1 \qquad L5 = 3.35$$

Thus, the limit moment for qualification loads is

$$mq := pw \cdot L5 - pb \cdot (L5 - L1) \qquad mq = 1.266 \times 10^{3}$$

Calculate section properties (refer to Fig. B-4):

$b1 := 6.00 \qquad tw1 := 0.20$

$a2 := 0.47 \qquad tw2 := 0.25$

$bh := 0.50 \qquad t3 := 0.100$

Area, A (in^2):

$$A1 := 2 \cdot a2 \cdot tw1$$

$$A2 := 2 \cdot a2 \cdot tw2 \qquad bnet := b1 - 3 \cdot bh$$

$$A3 := (bnet) \cdot t3 \qquad A := A1 + A2 + A3 \qquad \boxed{A = 0.873}$$

Centroid location, zb (in):

$$zb := \frac{1}{A} \cdot \left[A3 \cdot \frac{t3}{2} + A2 \cdot \left(t3 + \frac{a2}{2} \right) + A1 \cdot \left(t3 + \frac{a2}{2} \right) \right] \qquad zb = 0.188$$

Centroidal moment of inertia, Ix (in^4):

$$Ix1 := A3 \cdot \left(\frac{t3}{2} - zb \right)^2 + A2 \cdot \left(t3 + \frac{a2}{2} - zb \right)^2 + A1 \cdot \left(t3 + \frac{a2}{2} - zb \right)^2$$

$$Ix2 := \frac{1}{12} \cdot \left[(bnet) \cdot t3^3 + 2 \cdot tw2 \cdot a2^3 + 2 \cdot tw1 \cdot a2^3 \right]$$

$$Ix := Ix1 + Ix2 \qquad \boxed{Ix = 0.026}$$

Failure Mode 4: Bending of Baseplate Ribs (continued)

Calculate the limit bending stress (psi) in the rib section for qualification random vibration:

$$fbq := mq\cdot\frac{(a2 + t3 - zb)}{Ix} \qquad fbq = 1.869 \times 10^{4}$$

For flight:

$$fbf := .707\cdot fbq \qquad fbf = 1.321 \times 10^{4}$$

Allowable bending stresses:

Yield: $Fcy = 3.5 \times 10^{4}$

Ultimate: $Fbuf = 6 \times 10^{4}$

Margins of safety for qual testing:

$$MSyq4 := \frac{Fcy}{FSqy\cdot fbq} - 1 \qquad \boxed{MSyq4 = 0.873}$$

$$MSuq4 := \frac{Fbuf}{FSqu\cdot fbq} - 1 \qquad \boxed{MSuq4 = 1.568}$$

Margins of safety for flight:

$$MSyf4 := \frac{Fcy}{FSfy\cdot fbf} - 1 \qquad \boxed{MSyf4 = 1.408}$$

$$MSuf4 := \frac{Fbuf}{FSfu\cdot fbf} - 1 \qquad \boxed{MSuf4 = 2.244}$$

Note: These margins are based on a conservative assumption, which can be changed if necessary under the penalty of more detailed analysis. This assumption is that all of the moment introduced by the box's inertia is concentrated at the ends of the assumed beams. In reality, some of the moment is from the inertia of the baseplate itself, and some of the box's moment will be introduced from the other two panels, causing bending in the ribs running perpendicular to the ones assessed above.

Failure Mode 5: Shear in Bolts Attaching Box Walls to Baseplate

Allowable ultimate shear (lb) in fasteners, using the tensile-stress area, At, of the bolt because the bolts are into inserts and reducing the allowable by 25% to account for bolt bending: These bolts are #10s, so $At := 0.020$

$Psu := .75At \cdot Fsub$ $\qquad Psu = 1.425 \times 10^{3}$

Allowable bearing loads, Pbr (lbs), assuming a 25% reduction in strength because threads are in bearing:

Nominal thickness, $tnom := 0.10$ $\qquad$ Bolt diameter, $d := 0.190$
Minimum thickness, $tmin := 0.09$

Design thickness, $t := 1.1 \cdot tmin$ $\qquad t = 0.099$

Using the material properties on p. B-3:

$Pbry := .75 \cdot d \cdot t \cdot Fbry$ $\qquad Pbry = 705.375$

$Pbru := .75 \cdot d \cdot t \cdot Fbru$ $\qquad Pbru = 945.203$

Because Pbry<Psu, the joint is bearing critical, and failure should be ductile.

Assuming the load pw, calculated above for failure mode 4, will be shared equally by three bolts, the limit bolt shear for qual loads is

$$ps := \frac{pw}{3} \qquad ps = 235.501$$

Margins of safety for qual loads:

$$MSuq5a := \frac{Psu}{FSqu \cdot FF \cdot ps} - 1 \qquad \boxed{MSuq5a = 3.209}$$ (bolt shear, ult)

$$MSyq5b := \frac{Pbry}{FSqy \cdot ps} - 1 \qquad \boxed{MSyq5b = 1.995}$$ (bearing, yld)

$$MSuq5b := \frac{Pbru}{FSqu \cdot FF \cdot ps} - 1 \qquad \boxed{MSuq5b = 1.792}$$ (bearing, ult)

Margins of safety for flight loads:

$$MSuf5a := \frac{Psu}{FSfu \cdot FF \cdot 0.707ps} - 1 \qquad \boxed{MSuf5a = 4.316}$$ (bolt shear, ult)

$$MSyf5b := \frac{Pbry}{FSfy \cdot 0.707ps} - 1 \qquad \boxed{MSyf5b = 2.851}$$ (bearing, yld)

$$MSuf5b := \frac{Pbru}{FSfu \cdot FF \cdot 0.707 \cdot ps} - 1 \qquad \boxed{MSuf5b = 2.526}$$ (bearing, ult)

B.5 Assessment of Top Panel for Ground-Handling Loads

FalconSat-2 will be lifted from a lug attached to the top panel. The lug and panel will be tested to twice the limit load. The following analysis is an assessment of the top panel, based on factors of safety of 3 for ultimate and 2 for yield.

$$FSu := 3$$

$$FSy := 2$$

Limit load, p = total weight lifted (lb), as a concentrated load on the lug

$p := 100$ (accounting for the Pallet Ejection System and a fixture)

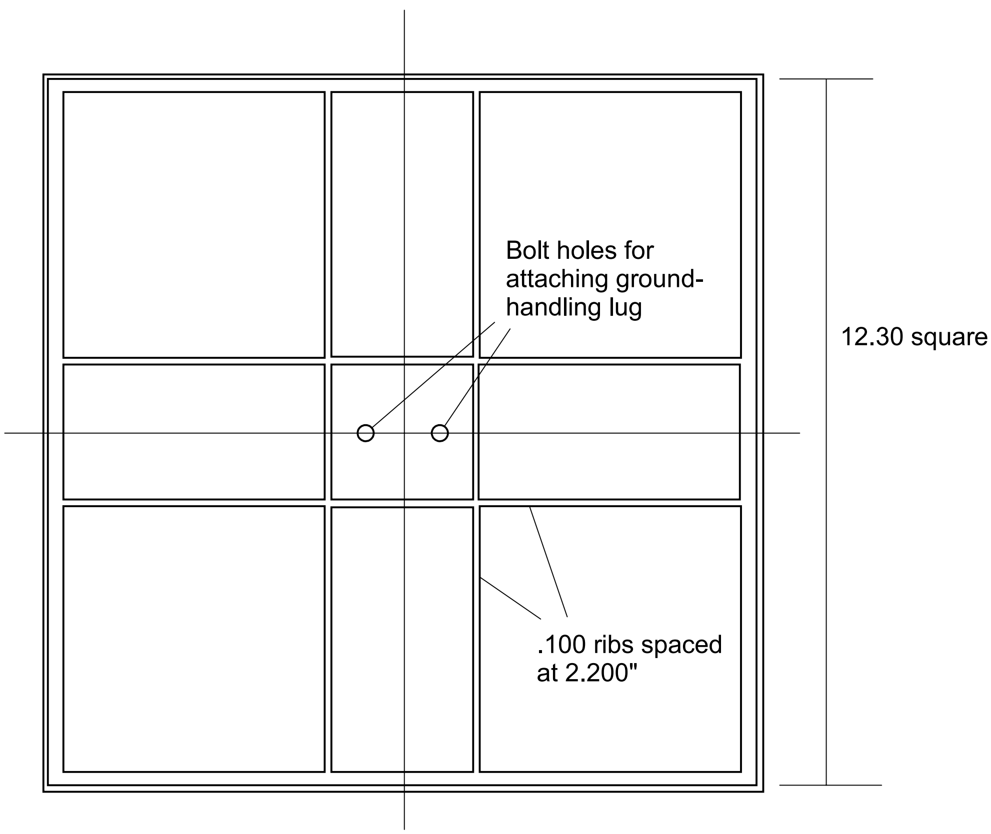
The lug is attached to the top panel by two #10 160-ksi bolts, with each carrying half the applied load.

The design ultimate and yield bolt loads (lb) are $ptu := FSu \cdot FF \cdot p$ $\quad ptu = 345$

$pty := FSy \cdot FF \cdot p$ $\quad pty = 230$

Two potential failure modes will be checked:

1. End pad bending near bolt hole from concentrated load
2. Bending of ribs in carrying the load out to the side panels



Top Panel for Ground-Handling Loads

Failure Mode 1: End Pad Bending

Analysis of a Tension Joint with a Single Bolt: Channel-Type Fitting
(equations derived from Lockheed Stress Memo #88a)

Prerequisite assumption and limitation (which is met):

- The fitting material is ductile (to be able to rely on the end-pad bending method)

Inputs:

$Db := .190$ (bolt diameter)

$At := .0200$ (tensile-stress area from above table)

$Dbh := .340$ (diameter of bolt head)

Calculated:

$$ri := \frac{Db}{2} \qquad ri = 0.095$$

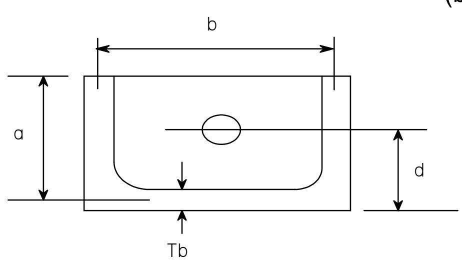
(bolt-hole radius)

$a := 1.05$ $\quad Tb := .100$

$b := 2.10$

$d := 0.55$ $\quad Te := 1.05\cdot(.100 - 0.0$

$Te = 0.095$



Analysis Process:

For the end-pad bending failure mode, calculate allowable bolt loads, which are the bolt loads that would cause the allowable stresses for that failure mode.

Compute margins of safety by comparing the calculated allowable bolt loads to the design bolt loads.

$$\frac{ri}{a} = 0.09$$ (Limitation: 0.1 < ri/a < 0.4)

$$\frac{b}{a} = 2$$ (Limitation: between 1.0 and 3.0)

$$K3 := -3.81 + \frac{3.8}{\left(\frac{ri}{a} + 0.015\right)^{.1}} + 0.15\cdot\ln\left(\frac{b}{a} - 0.30\right)$$

Note: The equation for K3 was developed in 1983 by Tom Sarafin to fit the curve used in Lockheed Stress Memo 88a. Staying within the above limitations, the calculated bending stress agrees with that calculated using the Stress Memo to within the accuracy possible by manually using the Memo curves.

Allowable ult. bolt tensile load for end-pad bending,

$$Ptu := \frac{\left(Fbuf\cdot a\cdot Te^2\right)}{K3\cdot(2d - Tb)}$$

$$Ptu = 547.243$$

$$MSu := \frac{Ptu}{ptu} - 1 \qquad \boxed{MSu = 0.586}$$

Top Panel for Ground-Handling Loads

Failure Mode 1: End Pad Bending (continued)

Allowable yield bolt tensile load for end-pad bending,

$$Pty := \frac{\left(Fcy \cdot a \cdot Te^2\right)}{K3 \cdot (2d - Tb)} \qquad Pty = 319.225$$

$$MSy := \frac{Pty}{pty} - 1 \qquad \boxed{MSy = 0.388}$$

Failure Mode 2: Bending of Ribs

Assume the applied load, p, is equally spread between the four side panels, transferred by shear and bending of the machined ribs. The ribs will be analyzed as simply supported beams.

Limit bending moment, m (in-lb), per rib:

$$m := \frac{p}{4(2)} \cdot \frac{12}{2} \qquad m = 75$$

As a conservative check, ignore the contribution of the skin.

Rectangular cross section of width, b, and thickness, t (in):

$$b := 0.100 \qquad t := 0.75$$

Limit bending stress, fb (psi):

$$fb := 6 \frac{m}{b \cdot t^2} \qquad fb = 8 \times 10^3$$

Margins of safety, using the same allowable stresses as for the base plate:

$$MSu := \frac{Fbuf}{FSu \cdot fb} - 1 \qquad \boxed{MSu = 1.5}$$

$$MSy := \frac{Fcy}{FSy \cdot fb} - 1 \qquad \boxed{MSy = 1.188}$$

B-6. Assessment of Joint Gapping at Limit Load

Bolt torque, T (in-lb), relates to preload as follows:

T = Kdp where K = nut factor
d = bolt diameter (in)
p = preload (lb)

For nonlubricated A-286 bolts with silver-plated A-286 nuts, K averages 0.2, but preload can vary about +/- 35% from nominal.

For A-286 bolts in nonlubricated 300-series SS spiralock inserts, assume a nominal value of K of 0.25, with preload varying +/- 35% from nominal.

Bolt preload does not contribute to bolt failure when the failure mode is ductile because, before the bolt fails, its material yields, and the joint gaps. This means that, at failure, the total bolt tensile load is equal to the applied load. The insert pull-out test performed at USAFA (see Appendix F) confirms that this is the case even when insert pull-out (thread stripping) is the critical failure mode, when the insert is installed in 6061-T6 aluminum alloy.

The preload (clamping force) must be high enough to ensure the joint does not gap at limit load. I will assume here that the joint would gap if the applied tensile load equals 1.2 times the preload, which is based on the assumption that the stiffness of the compressed load path is 5 times that of the bolt itself (a conservative assumption for aluminum fittings and steel bolts).

Bolts attaching base plate to ring (nuts):

Self-locking nuts, with run-in torque averaging about 12 in-lb (based on test at USAFA)

$Tr := 12$ This torque is ineffective in developing preload.

$K := 0.2$ $T := 100$ $d := 0.25$

$$p := \frac{T - Tr}{K \cdot d} \qquad p = 1.76 \times 10^{3}$$ -35%

$pmin := p \cdot (1 - .35)$ $pmin = 1.144 \times 10^{3}$

Flight limit tensile load (above):

$pt := .707 \cdot pt1$ $pt = 265.967$

Margin of safety for gapping:

$$MSg := \frac{pmin}{pt} - 1 \qquad \boxed{MSg = 3.301}$$

B-6. Assessment of Joint Gapping at Limit Load (continued)

#10 bolts attaching column to base plate and side panels to each other (inserts):

Spiralock inserts do not have a run-in torque but adequately lock the bolt.

$K := 0.25$ $\quad T := 60$ $\quad d := 0.19$

$$p := \frac{T}{K \cdot d} \qquad p = 1.263 \times 10^{3} \text{+/-35\%}$$

$$pmin := p \cdot (1 - .35) \qquad pmin = 821.053$$

Flight limit tensile load (for column attachment, above):

$$pt := .707 \cdot pt6 \qquad pt = 53.215$$

Margin of safety for gapping:

$$MSg := \frac{pmin}{pt} - 1 \qquad \boxed{MSg = 14.429}$$

#10 bolts attaching battery to base plate (nuts):

Self-locking nuts, with run-in torque assumed to average 10 in-lb

$Tr := 10$ This torque is ineffective in developing preload.

$K := 0.20$ $\quad T := 60$ $\quad d := 0.19$

$$p := \frac{T - Tr}{K \cdot d} \qquad p = 1.316 \times 10^{3} \text{+/-35\%}$$

$$pmin := p \cdot (1 - .35) \qquad pmin = 855.263$$

Flight limit tensile load (for battery attachment, above):

$$pt := .707 \cdot pt8 \qquad pt = 104.139$$

Margin of safety for gapping:

$$MSg := \frac{pmin}{pt} - 1 \qquad \boxed{MSg = 7.213}$$

FalconSat-2

Structural Verification Report

Appendix C

Finite-Element Model

Tom Sarafin

May 16, 2002

Contents:

C.1 Introduction

Several MSC/Nastran finite-element models of FalconSat-2 (FS2) were developed to help in design and verification. The main purpose of the models was to help us understand the dynamic characteristics, i.e., natural frequencies and modes of vibration. The models are simple, with about 1000 degrees of freedom (DOFs), and were intended to predict only the fundamental axial and lateral modes.

The first series of models represent the FalconSat-2 Engineering Model (EM). These models were generated immediately following the EM test program, which took place in April, 2001. I iterated these models with various modeling assumptions until the final version, FS2Emfix_4, accurately predicted the 182-Hz frequency of the fundamental lateral (rocking) mode, as determined by the EM test. This exercise gave me insight that later helped me generate a model that represented the configuration of the Qualification Model (QM).

We used our understanding of modal behavior and response to random vibration to design the structure for the QM. The first task was to generate design loads, based on predicted mass properties, the mode shape for the fundamental rocking mode, and the accelerations measured in EM testing. These design loads were eventually modified, as documented in Appendix A, based on updated mass properties and accelerations measured during QM testing.

The test-verified model of the QM is named FS2QM_4, and the current model of the flight unit is named FS2FM. FS2FM predicts a fundamental rocking-mode frequency of 139 Hz, as compared to 157 Hz measured in the QM test (and 153 Hz calculated for the FS2QM_4 model). The difference is because of two changes: (1) the separation switches bottomed out in the QM test because of an interference with the Pallet Ejection System provided by NASA, and (2) the base-plate thickness for the flight model was reduced by 0.180" in the center region to avoid a sep-switch interference during the mission.

Verification of structural requirements for FS2 is only secondarily dependent on the finite-element models. The stress analysis (Appendix B) is based on beam analogies and semi-empirical methods. The finite-element models were used to build understanding and support the assumptions made to generate design loads from test data.

C.2 Log of Model Runs and Key Results

Table C-1 summarizes the evolution and results of the FalconSat-2 finite-element models.

Table C-1. Log of Nastran Models. Models are grounded at the base.

Name	Date	Description	Results
FS2emmd	4/23/01	Crude model of the FS2 engineering model tested Apr, 2001. This model is grounded at the base of the ring. Modes run. Total weight = 48 lb. This model probably underpredicts the fundamental frequency because of the way the base plate is modeled.	Fundamental rocking frequency, fn = 114 Hz (actual from test was 182 Hz)
FS2emfix	4/23/01	FS2emmd with addition of the 0.75"-thick mounting plate, which is grounded along the edges (15" square), like in the test. Total weight = 67 lb.	fn = 101 Hz
FS2emfix_2	10/5/01	Completely revised model of the EM, with test mounting plate. The base plate was modeled a little more accurately in regard to the length of the beams with the most strain energy. Weight of all but mounting plate = 48.2 lb.	fn = 203 Hz
FS2emfix_3	10/12/01	FS2emfix_2 with dedicated nodes defined at the top of the ring. These nodes are then rigidly tied to the corresponding base plate nodes only in the 123 (translational) DOFs.	fn = 199 Hz
FS2EMfix4	10/22/01	FS2emfix_3 with modulus of elasticity dropped to 8.3E+6 to correlate fundamental frequency. This is the test-correlated EM model.	fn = 182 Hz
FS2EMfix4rv2	10/25	FS2Emfix4 configured for random vibration, with driven node rigidly attached to outer-edge grids of mounting plate. 3% damping. Run at CARS qual levels in X axis, between 50 and 400 Hz	RMS acceleration at center of top panel = 26.6 g (vs. 41.2 g in test), and peak at 41 g^2/Hz (vs. 56 in test)
FS2Emfix4rv3	10/25	FS2Emfix4rv2 with 1.3% damping.	RMS accel at top center = 40.5 (vs. 41.2 in test)
FS2QMx	10/8/01	New model corresponding to the design of the flight unit. Material densities were computed based on estimates only. Grounded at base of ring; no mounting plate. Weight = 44.7 lb.	fn = 149.8 Hz
FS2QM	10/10/01	FS2QMx with more detailed representation of base plate and interfaces with column and battery. Also has revised representations of	fn = 156.2 Hz (rocking at 45 degrees for the first time)

		column, battery, and sep switch stand. Top panel also revised slightly. Wt = 43.8 lb.	
FS2QM_2	10/12/01	FS2QM with dedicated nodes defined at the top of the ring. These nodes are then rigidly tied to the corresponding base plate nodes only in the 123 DOFs. Weight = 43.8 lb.	fn = 113.5 Hz, rocking at 45 degrees. For the first time, the column and battery are rocking opposite the box.
FS2QM_3a	10/16	FS2QM_2 with side-panel property (#4) modified such that 12I/t3 = 1000	fn1 = 163 Hz
FS2QM_3b	10/16	FS2QM_2 with side-panel property (#4) modified such that 12I/t3 = 10	fn1 = 121 Hz
FS2QM_2a	10/16	FS2QM_2 with revised model of switch stand. Weight = 43.6 lb.	fn1 = 113.5 Hz
FS2QM_2b	10/16	FS2QM_2a with pin releases at ends of machined ribs	fn1 = 112.9 Hz
FS2QM_2c	10/16	FS2QM_2b with .75"-deep ribs in top plate and side panels (was .50"). Weight = 44.4 lb.	fn1 = 111.9 Hz
FS2QM_3	10/16	FS2QM_2b with increased I for machined ribs to include effective skin. Weight = 44.4 lb. This is my best attempt to date for a structural model. Still needs revised for actual (or itemized predictions of) mass distribution. The side panels should be revised to include the stiffening effects of the solar panels. Also needs a representation of the test fixture.	fn1 = 123.5 Hz
FS2QM3rv6	10/25	FS2QM_3 configured for random vibe analysis, with base-driven point centered at base of ring and rigidly attached to ring grids. Damping input at 3%. Run in the X axis at CARS qual levels from 50 to 300 Hz.	RMS acceleration at the center of the top panel = 17.3 g
FS2QM3rv7	10/25	FS2QM3rv6 with 1.3% damping, which is the value that provided good correlation with RMS response acceleration for the EM. This is the best prediction to date of random-vibration response for the qual model.	RMS accel at top center = 26.2 g. RMS bending stress in base plate ribs = 8.8 ksi
FS2QM_4a	5/2/02	FS2QM_3 with the following changes: - Higher-fidelity model of switch plate - +/- X side panels with different mass than +/- Y panels to reflect difference	fn = 123.2 Hz

		in solar arrays - Mass properties revised to agree with MP report 8.0 (43.1 lb)	
FS2QM_4b	5/2/02	FS2QM_4a with ends of sep switches grounded in DOFs 1, 2, and 3.	fn = 128.7 Hz
FS2QM_4c	5/2/02	FS2QM_4b with switch bracket plates modeled at 1" thickness, with density reduced proportionally. This is intended as the upper bound for the effect of switch grounding.	fn = 137.9 Hz
FS2QM_4d	5/2/02	FS2QM_4b with pin releases removed from bars representing machined ribs (releases first used in FS2QM_2b)	fn = 132.7 Hz
FS2QM_4e	5/3/02	FS2QM_4d with the following changes: - Corrected modeling of interior base plate ribs (grids 145, 154, 156, and 165 had previously never been connected by bar elements) - Increased moments of inertia for bar elements representing the base plate ribs.	fn1 = 147.8 Hz fn2 = 151.4 Hz
FS2QM_4	5/3/02	FS2QM_4e with side-panel ribs stiffened to I = 0.02 in4. This is the test-correlated model of the qualification test configuration. Weight = 43.06 lb.	fn1 = 153.1 Hz fn2 = 156.5 Hz (vs. 157 and 171 in test)
FS2Fma	5/3/02	FS2QM_4 with sep switches unconstrained	fn1 = 148.1 Hz
FS2FMb	5/3/02	FS2FMa with spring elements to represent tips of sep switches	fn1 = 148.1 Hz
FS2FM	5/3/02	FS2FMb with I of internal ribs in base plate reduced to reflect 0.180" reduction in thickness. This model reflects the flight configuration.	fn1 = 139.1 Hz

C.3 Model Plots

Figures C-1 shows the undeformed FS2FM model. Figures C-2 and C-3 show the fundamental rocking mode and the fundamental axial mode predicted by the FS2FM model.

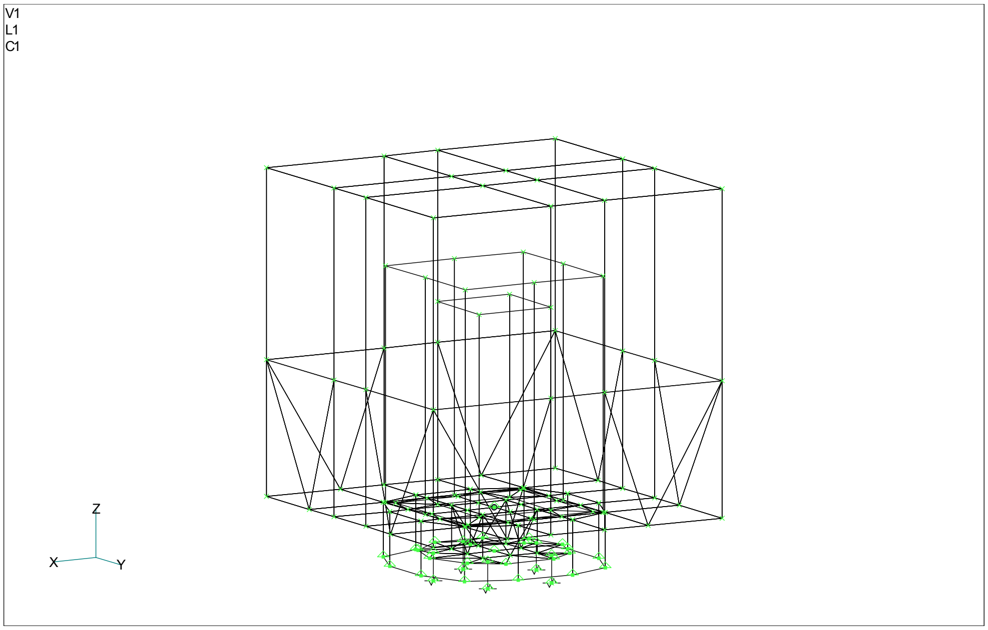


Fig. C-1. Undeformed Plot, FS2FM.

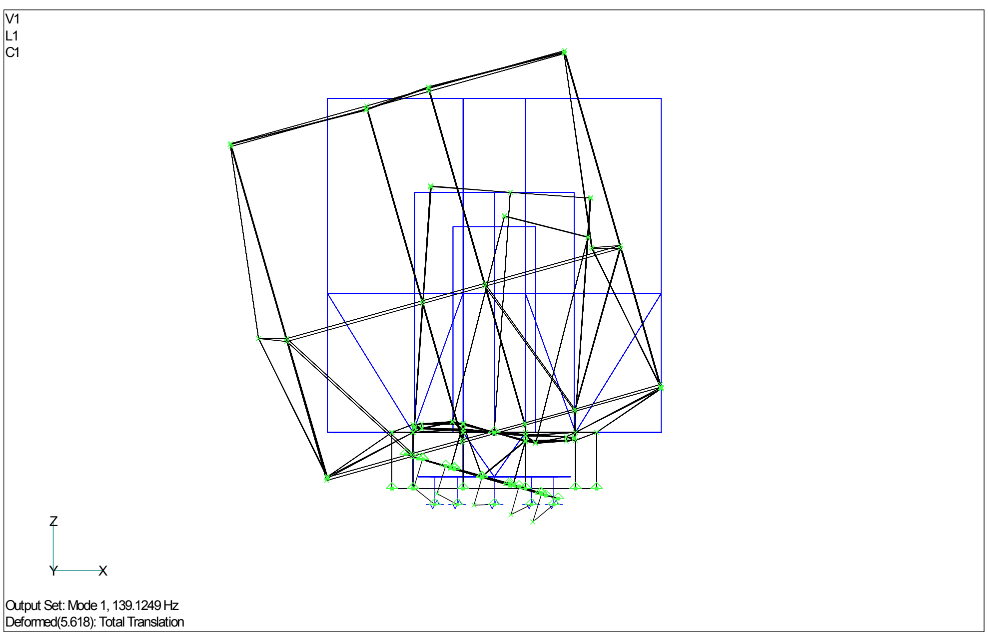


Fig. C-2. Fundamental Rocking Mode, FS2FM.

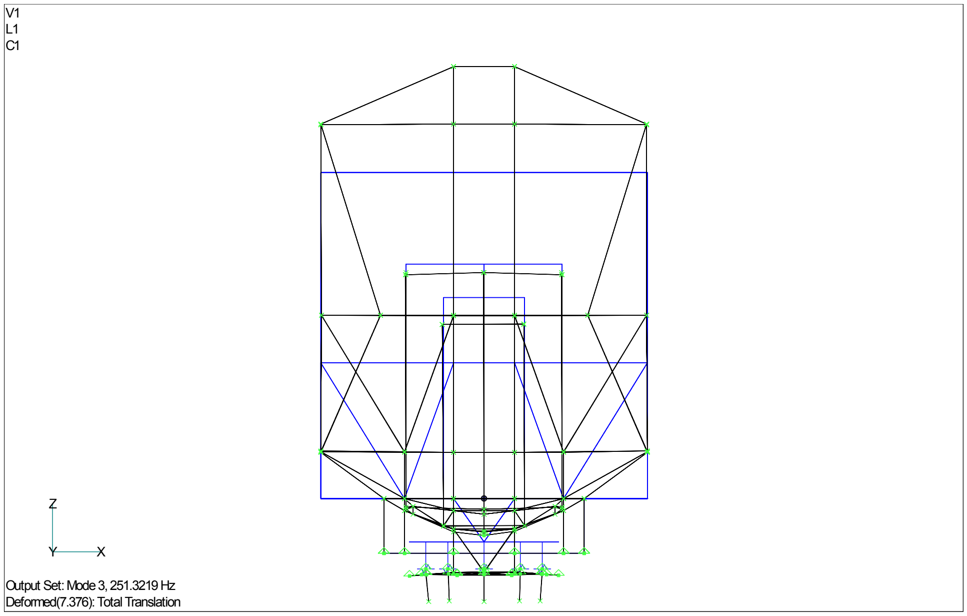


Fig. C-3. Fundamental Axial Mode, FS2FM.

C.4 Mass Properties Comparison

Table C-2 compares mass properties of the FS2QM_4 model with Version 8.0 of the mass properties report for the FS2 qualification model. (See Appendix H.) As a comparison of Appendix H and Appendix I (mass properties for flight model) shows, there is little difference between the mass properties of the QM and the FM. Also, there is no difference between mass properties of the FS2QM_4 and FS2FM models. Thus, the FS2FM model mass properties were not revised to agree with Appendix I.

Table C-3 is a summary of mass properties for the FS2FM model, as calculated by Nastran.

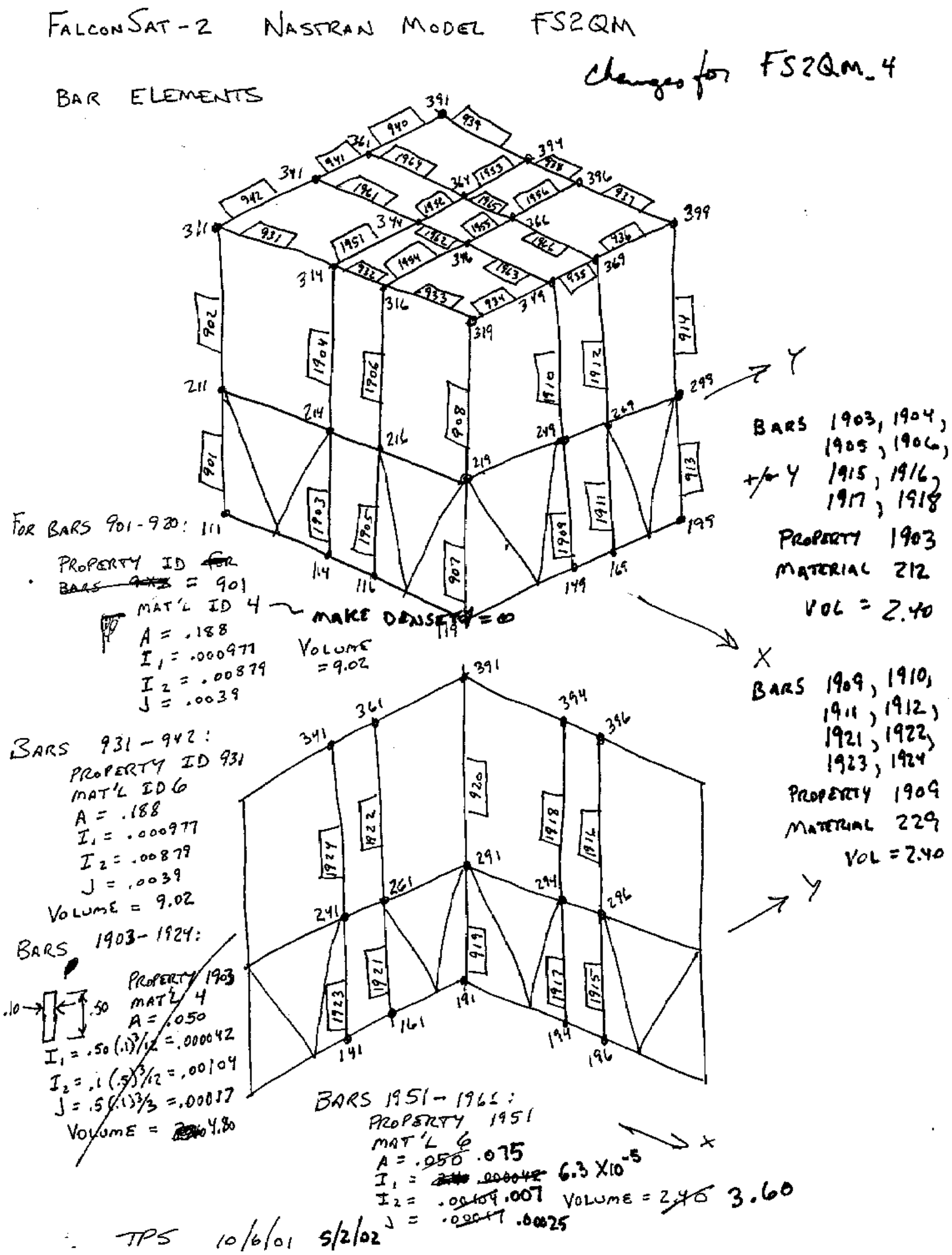
Table C-2. Comparison of Mass Properties: QM Mass Properties V8.0 versus the FS2QM_4 Model. The "weight" column on the left-hand side is from the mass-properties report. The spreadsheet was used to calculate target material densities for the model. Values in the "Actual Model Density" column are those used for the model. The total error in mass is negligible.

Description	Weight (lb)	Mass (lb-s2/in)	Property ID	Property Volume (in3)	Material ID	Material Volume (in3)	Target Density (lb-s2/in4)	Actual Model Density (lb-s2/in4)	Model Weight (lb)	Error	% Error
Base plate		0	1	3.80							
Base plate		0	2	18.00							
Base plate		0	3	4.30							
Base plate		0	701 & 706	25.20							
Base plate total	4.56	0.01181			1	51.30	0.0002302	0.000230	4.56	0.00	0%
X side panels (2)		0	229	28.80							
X side panels (2)		0	1909	2.40							
Side panels total	8.50	0.022015			229	31.20	0.0007056	0.000706	8.50	0.00	0%
Y side panels (2)		0	212	28.80							
Y side panels (2)		0	1903	2.40							
Side panels total	5.83	0.0151			212	31.20	0.000484	0.000484	5.83	0.00	0%
Top panel		0	5	14.30							
Top panel		0	6	4.40							
Top panel		0	455	0.50							
Top panel		0	931	9.00							
Top panel		0	1951	3.60							
Top panel total	4.72	0.012225				31.80	0.0003844		4.71	-0.01	0%
					5	14.30	0.0003844	0.000384	2.12		
					6	17.00	0.0003844	0.000384	2.52		
					455	0.50	0.0003844	0.000384	0.07		
Adapter ring	3.00	0.00777	7	49.20	7	49.20	0.0001579	0.000158	3.00	0.00	0%
Column	10.46	0.027091	8	51.10	8	51.10	0.0005302	0.000530	10.46	0.00	0%
Battery	4.32	0.011189	1101	44.40	1101	44.40	0.000252	0.000252	4.32	0.00	0%
Sep switch stand	0.69	0.001787	1501	7.60	1501	7.60	0.0002351	0.000235	0.69	0.00	0%
Sep switches	1.03	0.002668	561	3.75	561	3.75	0.0007114	0.000711	1.03	0.00	0%
Total FS2	**43.11**	**0.111655**							**43.10**	-0.01	0%

Table C-3. Mass Properties Calculated for the FS2FM Model. The origin of the model coordinate system is at the center of the base plate, centered also through the 0.75" thickness. For comparison, the QM mass properties report, V.8.0, shows a center of gravity at X = 0.039", Y = -0.030", and Z = 5.055". The origin of the coordinate system used in the mass properties report is centered at the aft end of the base plate, at the mating interface with the separation ring. This origin is 0.375" aft of the one used in the model, so, for an apples-to-apples comparison, add 0.375" to the 4.558" c.g. location for the model: 0.375 + 4.558 = 4.933" vs. 5.055" in the mass properties report. For the FM mass properties report (flight model), the c.g. location is X = -0.004, Y = -0.020, and Z = 4.894.

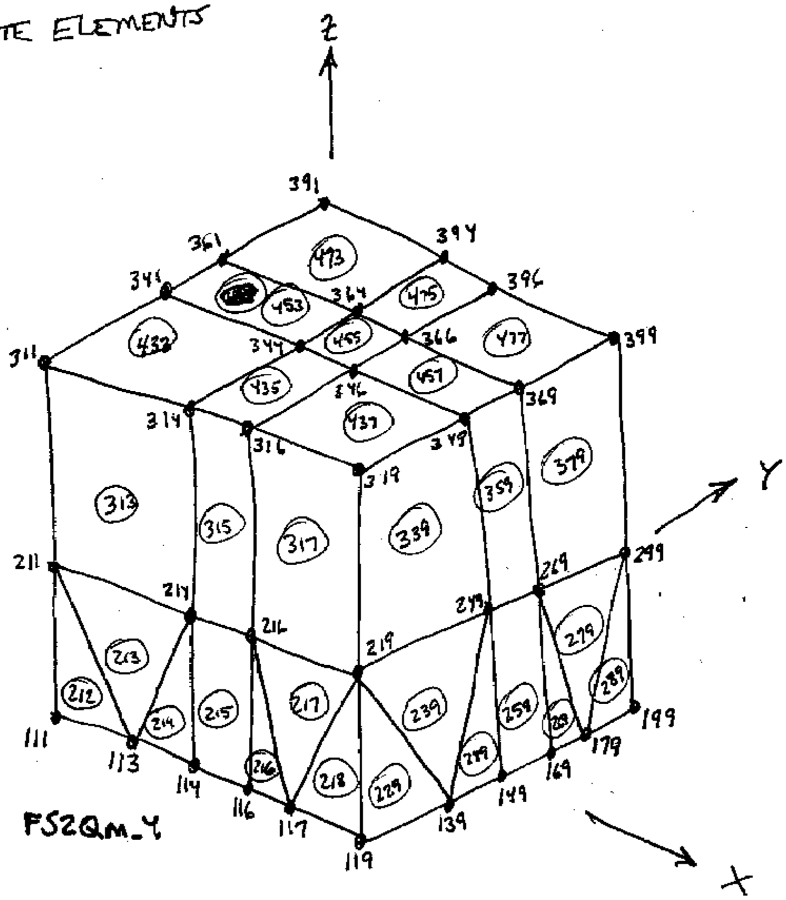
```
Tools Mass Properties Mesh Properties
335 Element(s) Selected...

Mass                    Center of Gravity in CSys 0
Structural   =    0.11154  X=   3.35693E-4  Y=   -3.7406E-4  Z=    4.558231
NonStructural=         0.  X=           0.  Y=           0.  Z=          0.
Total Mass   =    0.11154  X=   3.35693E-4  Y=   -3.7406E-4  Z=    4.558231

Inertias about CSys 0                 Inertias about C.G. in CSys 0
Ixx =    5.387463  Ixy=  9.36431E-5    Ixx =    3.069939  Ixy=  9.36571E-5
Iyy =    5.567528  Iyz=  -1.1057E-5    Iyy =    3.250003  Iyz=  1.79127E-4
Izz =    2.772695  Izx=          0.    Izz =    2.772695  Izx=  -1.7068E-4

Total Length (Line Elements only) =        389.
Total Area   (Area Elements only) =    1242.217
Total Volume (All Elements)       =    310.1948
```

C.5 Model Definition



FalconSat-2 Nastran Model FS2QM

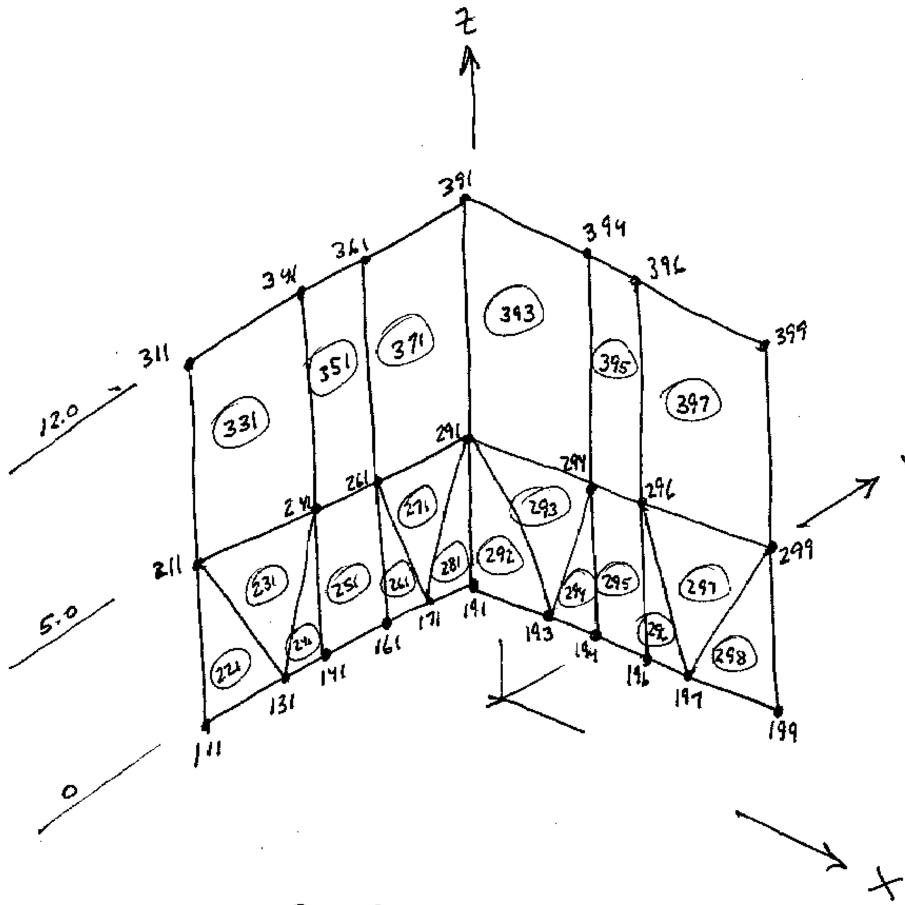
PLATE ELEMENTS



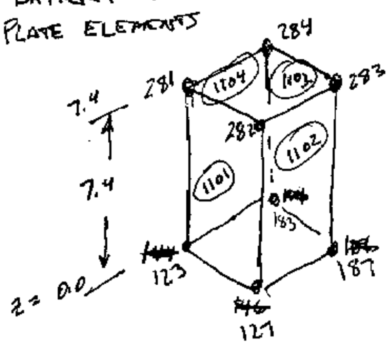
changes for FS2QM_Y

ELEMENTS	PROPERTY ID	MAT'L ID	THICKNESS	VOLUME
SIDE WALLS: 2XX, 3XX	~~4~~	~~4~~	.100	57.6
TOP PANEL: 433, 437, 473, 477	5	5	.150	14.3
435, 453, ~~455~~, 457, 475	6	6	.100	~~4.9~~ 4.4
455	455	455	.100	0.57
+/- Y PANELS	212	212	.100	28.8
TPS 10/5/01 5/2/02 +/- X PANELS	229	229	.100	28.8

FALCON SAT-2 NASTRAN MODEL FS2QM



BATTERY PACK:
PLATE ELEMENTS

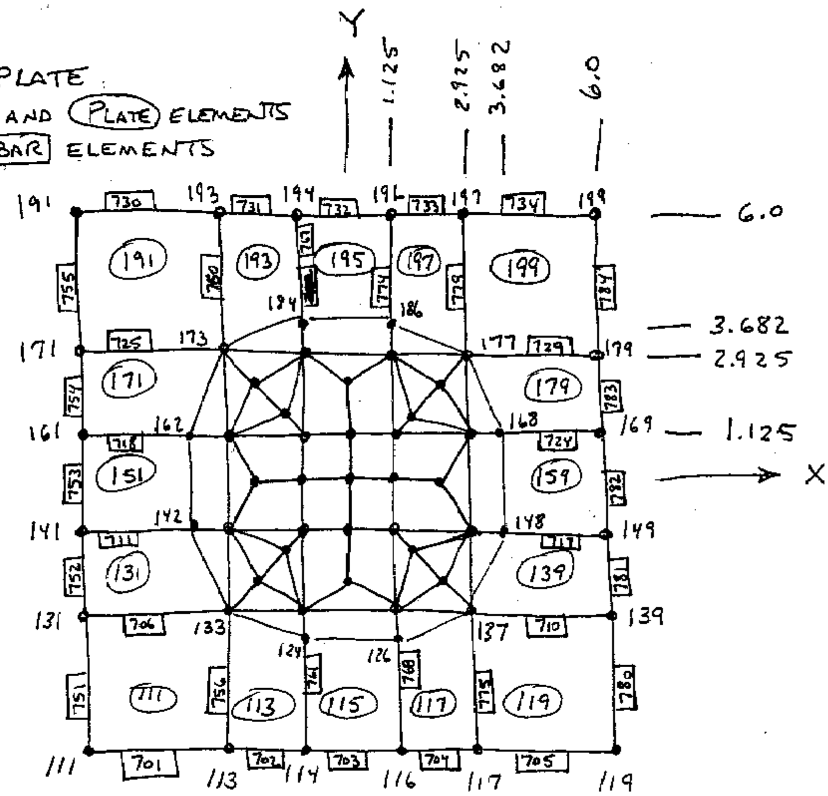


PROPERTY ID 1101
MATERIAL 1101 $E = 29 \times 10^6$
t = .50
Volume = ~~33.3~~ 44.4
~~Wt = 3.58~~

TPS 10/05/01

FALCON SAT-2 NASTRAN MODEL FS2QM

BASEPLATE
GRIDS AND (PLATE) ELEMENTS
AND [BAR] ELEMENTS



ELEMENTS	PROPERTY	MATERIAL	THICKNESS	VOLUME
111, 119, 191, 199	1	1	.100	3.8
ALL OTHER PLATES THIS VIEW	2	1	.250	18.0

BARS 706, 710, 725, 729,
756, 760, 775, 779
PROPERTY ID 706
MATERIAL ID 1

.20 × .75

$A = .20(.75) = .150$
$I_1 = .75(.20)^3/12 = .00050$
$I_2 = .20(.75)^3/12 = .00703$
$J = .75(.20)^3/3 = .0020$

VOLUME = 25.2
TOTAL FOR ALL 701 & 706

ALL OTHER BARS
THIS VIEW
PROPERTY ID 701
MATERIAL ID 1

.25 × .75

$A = .25(.75) = .188$
$I_1 = .75(.25)^3/12 = .000977$
$I_2 = .25(.75)^3/12 = .00879$
$J = .75(.25)^3/12 = .0039$

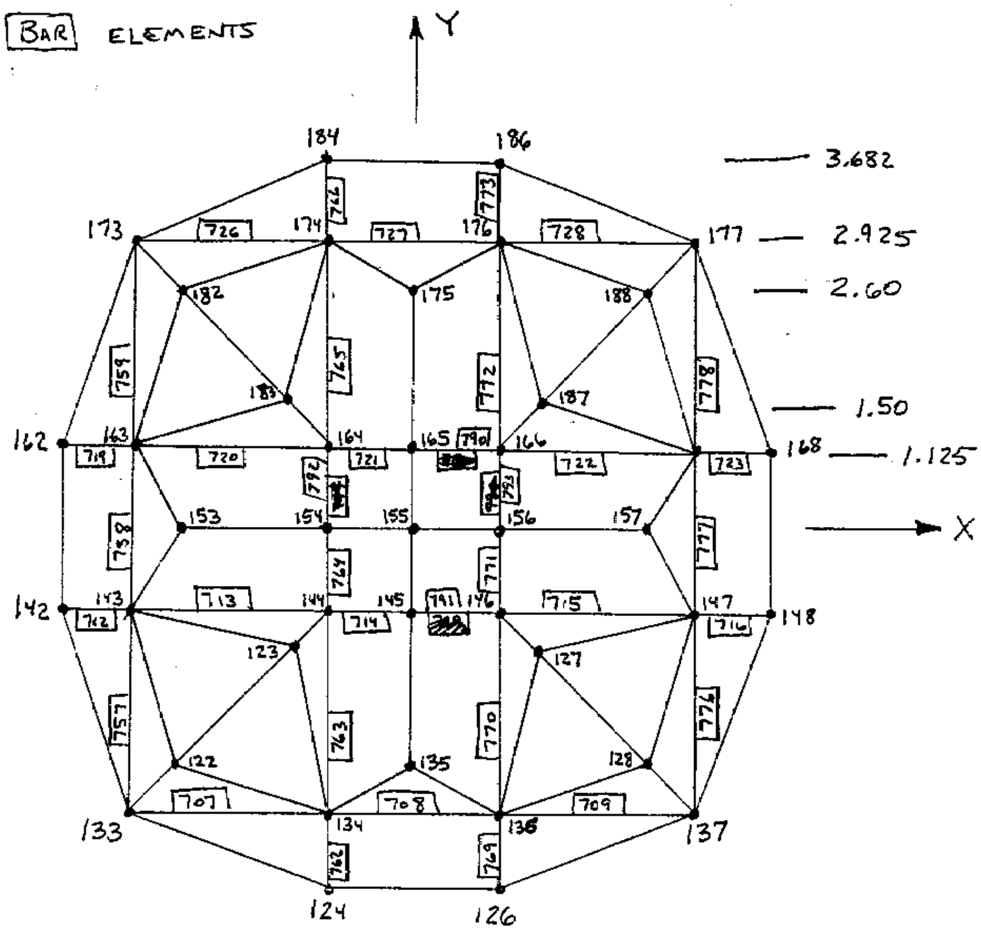
~~VOLUME~~

TPSARAFIN 10/8/01

FALCONSAT-2 NASTRAN MODEL FS2QM

CENTER PORTION OF BASEPLATE

$\boxed{\text{BAR}}$ ELEMENTS



BARS 707, 708, 709, 726, 727, 728 } PROPERTY ID 706
757, 758, 759, 776, 777, 778 }

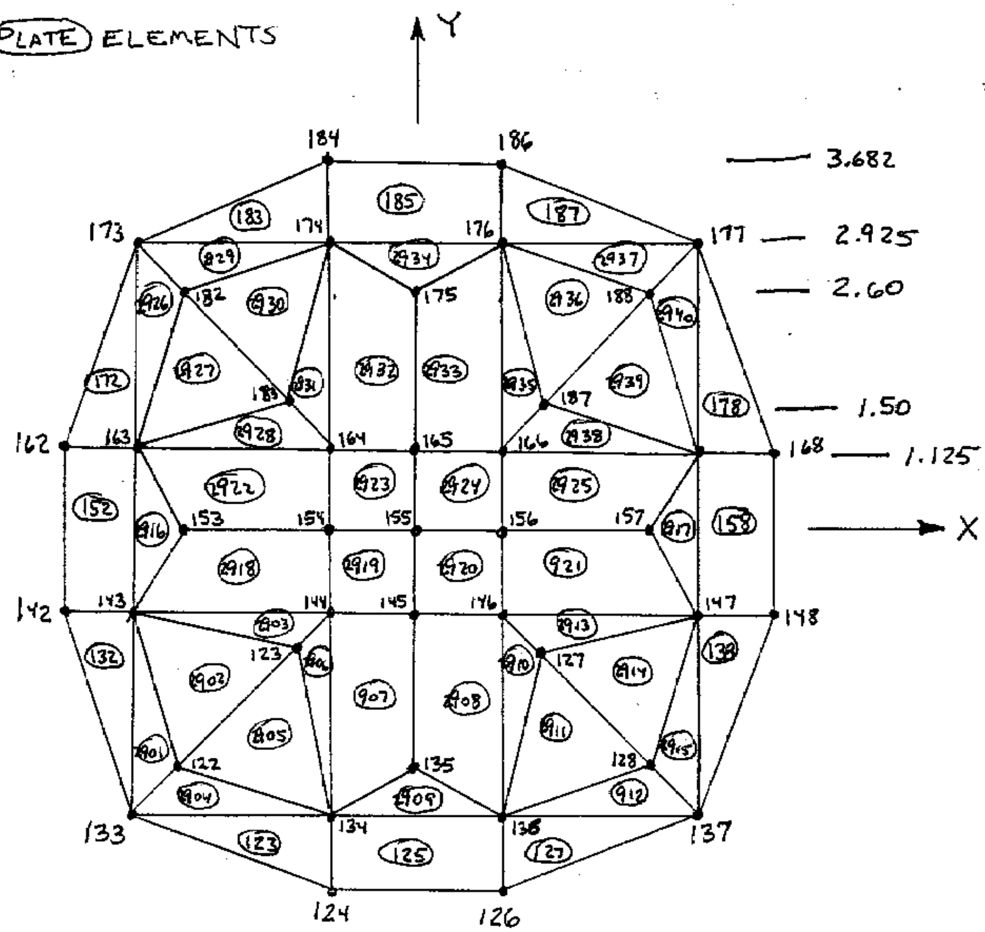
ALL OTHER BARS THIS VIEW: PROPERTY ID 701

TPSARAFIN 10/8/01

FALCONSAT-2 NASTRAN MODEL FS2QM

CENTER PORTION OF BASEPLATE

(PLATE) ELEMENTS



PLATES 9XX : PROPERTY ID 3
MATERIAL ID 1
t = .125

VOLUME = 4.3

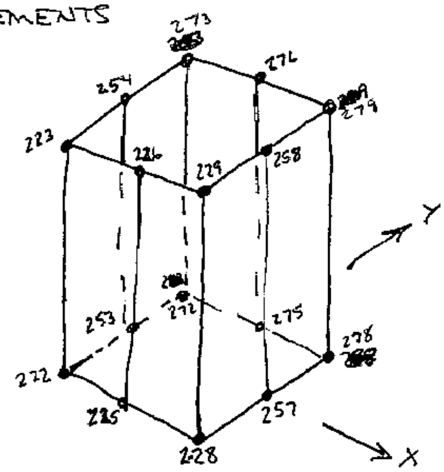
ALL OTHER PLATES THIS VIEW:
PROPERTY ID 2
MATERIAL ID 1
t = .250

TPSARAFIN 10/8/01

FALCONSAT-2 NASTRAN MODEL FS2QM

COLUMN

PLATE ELEMENTS

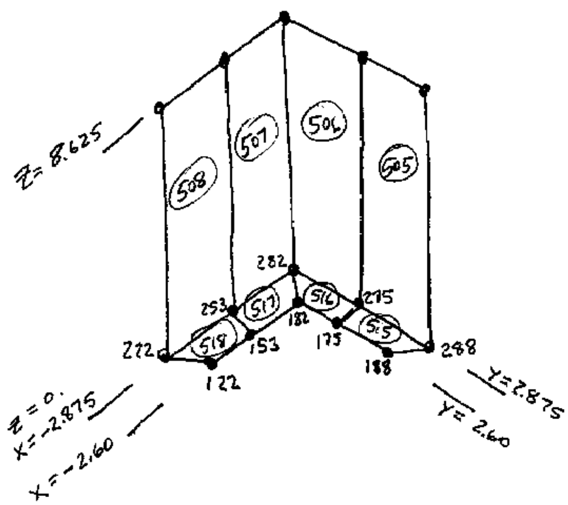


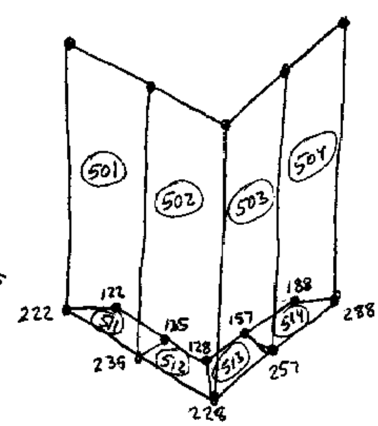
PROPERTY ID 8

MATERIAL ID 8

t = .250

VOLUME = 51.1





TPSARAFIN 10/8/01

FALCONSAT-2 NASTRAN MODEL FS2QM

~~Changes~~ CHANGES FOR FS2QM_2a

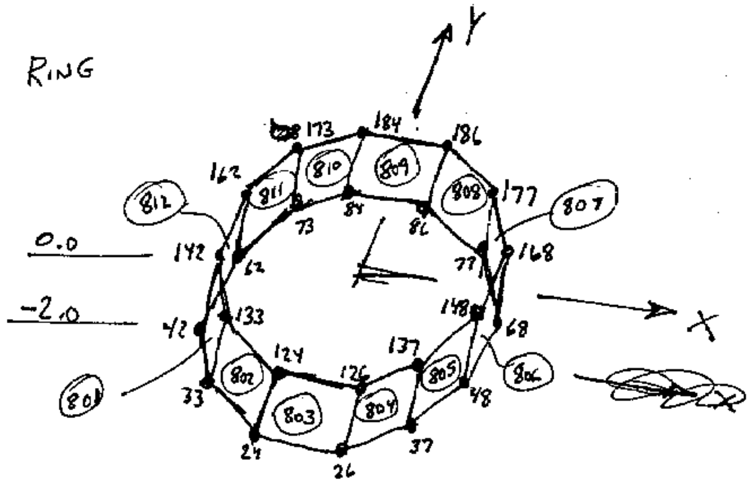
RING

PROPERTY ID 7

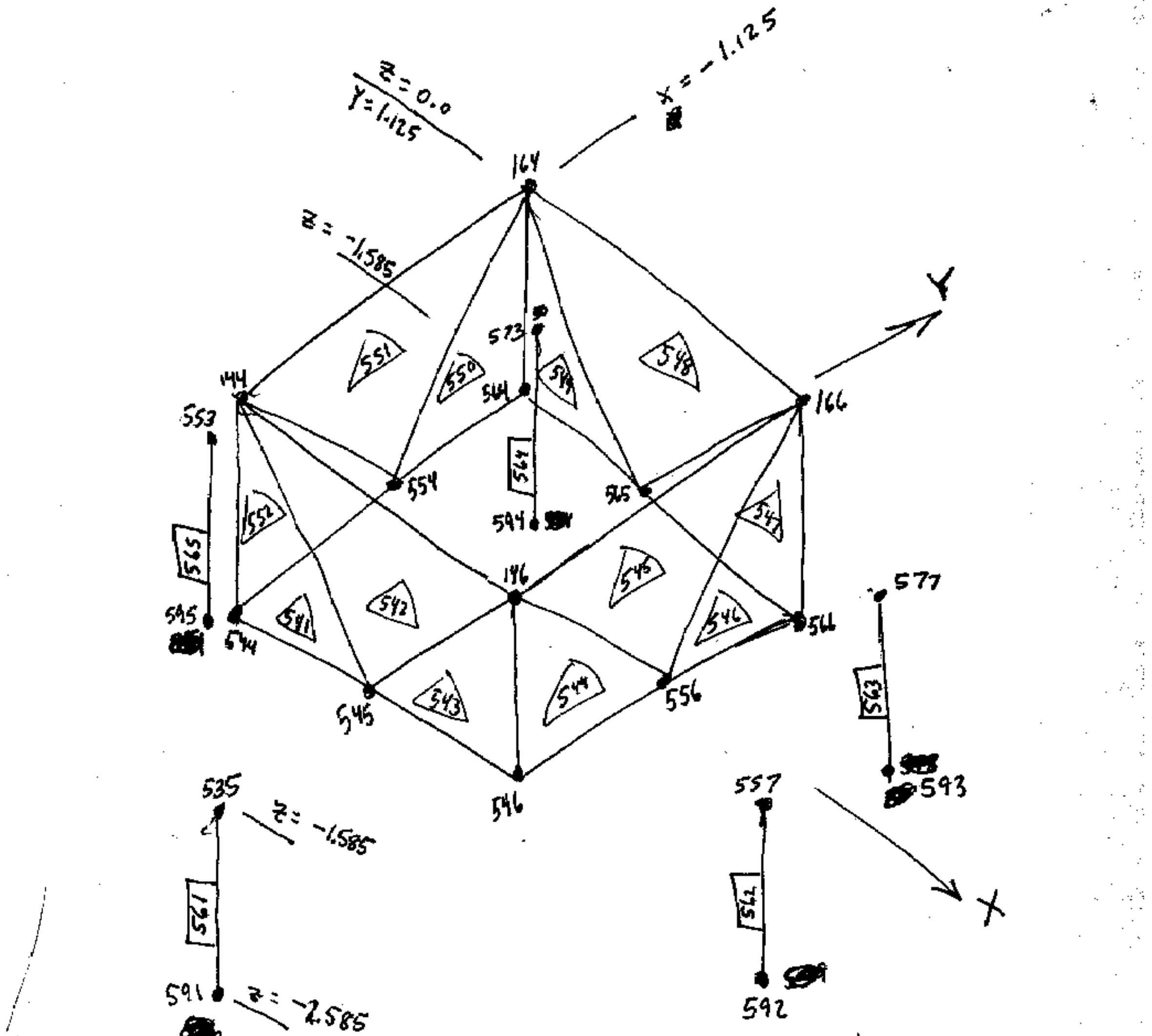
MATERIAL ID 7

$t = 1.0$

VOLUME = 49.2



FALCONSAT-2 NASTRAN MODEL
SEP-SWITCH BRACKET & SWITCHES

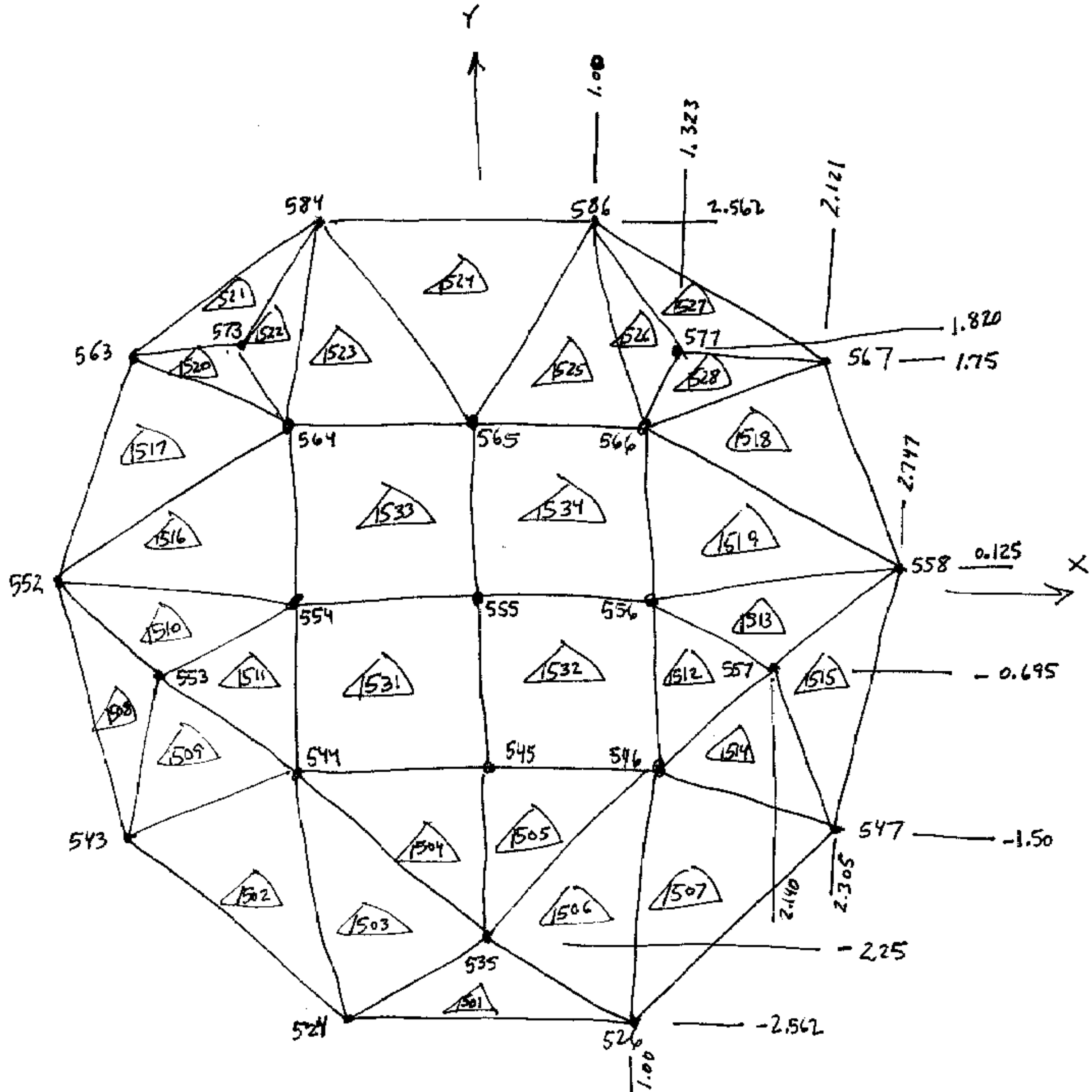


PLATES:
PROPERTY ID 1501
MATERIAL ID 1501
$t = .200$

BARS (SWITCHES):
PROPERTY 561
MATERIAL 561
$A = .75 \text{ in}^2$
$I_1 = I_2 = J = .10 \text{ in}^4$
$\text{Volume} = 3.75 \text{ in}^3$

TPSARAFIN 5/1/02

FALCONSAT-2 NASTRAN MODEL
SEP SWITCH PLATE



ALL PLATES IN THIS VIEW:
PROPERTY ID 1501
MATERIAL ID 1501
$t = .200$

VOLUME = 7.60 in^3

TPSARAFIN 5/1/02

FS2FM NASTRAN MODEL — FLIGHT MODEL

SPRINGS FOR SEP SWITCHES

$$K \simeq \frac{16\ \text{LB}}{.30\ \text{IN}} = 53\ \text{LB/IN} \quad \text{PER SWITCH}$$

AS MEASURED

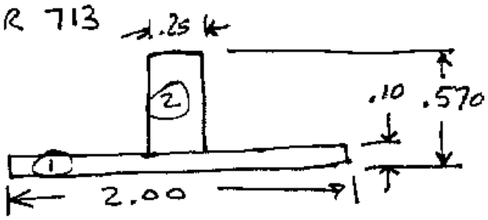
ADD SPRING ELEMENTS (DOF SPRINGS ~~CELAS2~~) TO GRIDS 591 – 595
IN THE Z DIRECTION, K = 53 LB/IN

ELEMENTS 591 – 595

ADD GRIDS 596 – 600 coincident

INTERIOR RIBS IN BASEPLATE (MACHINED DOWN BY 0.180" TO ACCOMMODATE THE SWITCH BRACKET)

PBAR 713



SEC	A	z'	Az'	z	Az^2	I_o
1	.200	.050	.0100	−.1058	.00224	.00017
2	.118	.335	.0395	.1792	.00379	.00216
Σ	.318	.1558	.0495		.00603	.00233

$I = .00603 + .00233 = .00836 \text{ IN}^4$

AS MODELED:

PLATE $I = \dfrac{2(.10)^3}{12} = .00017 \text{ IN}^4$

BAR $I = .00836 - .00017 = \cancel{.00819}$ $.0082 \text{ IN}^4$

$J = .003 \text{ IN}^4$ (WAS .0039)

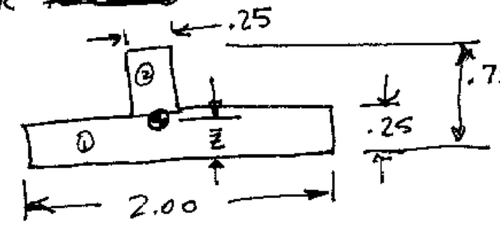
TPSARAFIN 5/3/02

FALCONSAT-2 NASTRAN MODEL
MODEL FS2QM_4

BASEPLATE RIBS

FOR CBARS 711, 712, 716, 717
718, 719, 723, 724
761, 762, 766, 767
768, 769, 773, 774

PBAR [redacted]
711



SEC	A	z'	Az'	z	Az^2	I_o
1	.500	.125	.0625	-.075	.00281	.00260
2	.125	.500	.0625	.300	.01125	.00260
Σ	.625	.200	.1250		.01406	.00521

$\bar{z}$ ↗ (.200)

$I = .01406 + .00521 = .0193 \text{ IN}^4$

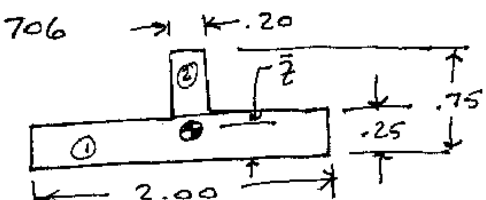
AS MODELED:

PLATE ELEMENT $I = \dfrac{2(.25)^3}{12} = .00260 \text{ IN}^4$ $A = .188 \text{ IN}^2$

BAR ELEMENT $I = .0193 - .00260 = \boxed{.0167 \text{ IN}^4}$

PBAR 706

FOR CBARS 706 - 710
725 - 729
756 - 760
775 - 779



SEC	A	z'	Az'	z	Az^2	I_o
1	.500	.125	.0625	-.0625	.00195	.00260
2	.100	.500	.0500	.3125	.00977	.00208
Σ	.600	.1875	.1125		.01172	.00414

$I = .01172 + .00414 = .0159 \text{ IN}^4$

AS MODELED:

PLATE ELEMENT $I = .00260 \text{ IN}^4$

BAR ELEMENT $I = .0159 - .00260 = \boxed{.0133 \text{ IN}^4}$

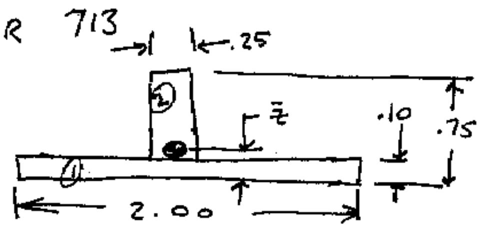
$A = .15 \text{ IN}^2$

TPSarafin 5/3/02

FALCONSAT-2 NASTRAN MODEL
MODEL FS2QM_4

BASEPLATE RIBS — FOR CBARS 713, 714, 715, 790
720, 721, 722, 791
763, 764, 765, 792
770, 771, 772, 793

PBAR 713



SEC	A	z'	Az'	$\bar{z}$	Az^2	I_o
1	.200	.050	.0100	−.1684	.00567	.00017
2	.163	.425	.0693	.2066	.00696	.00572
Σ	.363	.2184	.0793		.01263	.00589

$I = .01263 + .00589 = .0185 \text{ in}^4$

AS MODELED:

PLATE $I = \dfrac{2(.10)^3}{12} = .00017 \text{ in}^4$

BAR $I = .0185 - .00017 = \boxed{.0183 \text{ in}^4}$

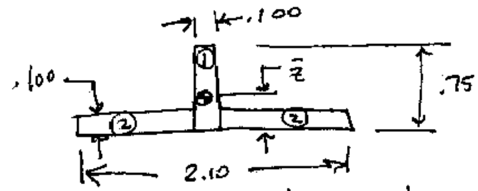
PBAR 701 — FOR CBARS 701 – 705
730 – 734
751 – 755
780 – 784

$A = .188 \text{ in}^2$

$I = .012 \text{ in}^4$

MODEL ~~FS2QM_3~~ FS2QM_4

SIDE PANEL RIBS



SEC	A	z'	Az'	$\bar{z}$	$A\bar{z}^2$	I_o
1	.075	.375	.0281	.236	.00418	.00352
2	.200	.050	.0100	−.089	.00158	.000167
Σ	.275	.139	.0381		.00576	.00368

$I = .00576 + .00368 = .00944$

$I_{3\,RIBS} = 3 \times I = .0283\ \text{in}^4$

AS MODELED (2 RIBS)

PLATE $I =$ [illegible] $\cancel{2(.00422)} + \frac{2}{\cancel{3}}(.000167) = \cancel{.00844}\ \text{in}^4$ $.00033$

FOR PBAR 1903, & 1909 increase I to $I \approx \frac{1}{\cancel{2}}\cancel{\frac{2}{3}}(.0283 - \cancel{3(.000167)}$ $.00033) = \cancel{.00927}$ $.014\ \text{in}^4$

~~$I = .009\ \text{in}^4$~~

USE $I = .02\ \text{in}^4$ to account for stiffening from solar arrays

MODEL RESULTS:

MASS = .115095

WT = 44.4 LB

f_{n1} = 123.4 HZ (ROCKING)

f_{n5} = 209 HZ (AXIAL)

~~f_{n4} = 269 (SIDES)~~

f_{n8} = 276 (TOP PANEL)

f_{n9} = 276 (sides)

TPS ~~10/16/01~~ 5/3/02

FalconSat-2

Structural Verification Report

Appendix D

Justification for Notching the Random Vibration Environment

Don Hershfeld

February, 2002

Vibration test specifications are generally based on free interface acceleration spectra and do not account for any influence of the attached payload. Scharton et al. have derived a theoretical basis for limiting vibration test levels using Norton's and Thevinin's equivalence theorems. The basis for limiting or notching vibration test levels is expressed by equation 1:

$$\frac{A}{A_0} + \frac{F}{F_0} = 1 \qquad (1)$$

where A_0 is the free interface acceleration and F_0 is the blocked force. The blocked force can be computed as the product of A_0 and the effective interface impedance. In the case of random vibration, A_0, F_0 are represented by power spectral densities. The blocked force spectral density is then computed as the product of A_0 and the square of the effective interface impedance.

For sinusoidal testing, the interface force is expressed as a scaled value of A. $F=Z_{P/L}*A$. The notched input acceleration that satisfies equation 1 can then be calculated:

$$\frac{A}{A_0} + \frac{Z_{P/L} * A}{Z_{I/F} * A_0} = 1$$

$$\left[1 + \frac{Z_{P/L}}{Z_{I/F}}\right] A = A_0 \qquad (2)$$

$$A = \left[\frac{Z_{I/F}}{Z_{I/F} + Z_{P/L}}\right] A_0$$

The magnitude of the notch can be calculated, in dB as:

$$MagnitudeofNotch(dB) = 20 * \log\left[\frac{Z_{I/F}}{Z_{I/F} + Z_{P/L}}\right] \qquad (3)$$

Note that, in the case of random testing, the bracketed impedance terms are squared. However, the magnitude of the notch, expressed in dB, remains the same. After computing the notching factor, it is a simple process to compute the reduced input acceleration and the maximum or limiting interface force.

In the case of Hitchhiker-PES the limiting load is the interface moment. Equation 1 can be extended for this case as shown:

$$\frac{A}{A_0} + \frac{M}{M_0} = 1 \qquad (4)$$

The magnitude of the notch relationship, given in equation 3, remains the same. Only the impedance terms are now moment impedances instead of force impedances.

In order to establish the notching criteria for Hitchhiker-PES payloads it is necessary to determine both the impedance of the PES interface and the impedance of the payload. These impedances may be computed using either finite element models (FEMs) or Craig-Bampton (C-B) models.

The interface impedance of the Hitchhiker-PES was computed using the C-B model provided by Chris Fransen (Swales) in SAI-TM-1977. Figure D-1 shows the interface impedances of the Hitchhiker-PES payload interface as the moment required at the interface to produce a 1 g lateral acceleration. The impedances have been computed for both lateral axes. The maximum curve in this figure represents the maximum of the X and Y axis impedances plus a margin of 3 dB. This maximum impedance is used later to compute the Falconsat notch.

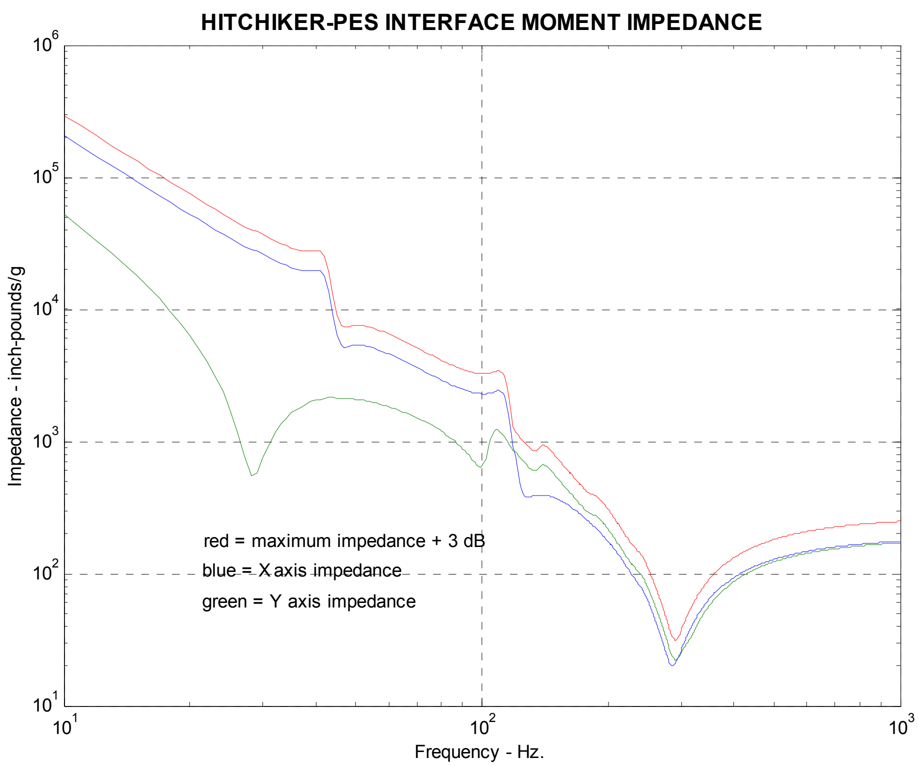


Fig. D-1. Hitchhiker-PES Interface Moment Impedance.

A Craig-Bampton model of Falconsat-2 has been created based on a Simplified Falconsat-2 Math Model for Fundamental Lateral Vibration, provided by Tom Sarafin. The model consists of a 25 pound weight and 2.5 lb-in-sec^2 mass moment of inertia attached to a 6 inch long rigid bar element. The rigid bar element is constrained at its base by a torsion spring element having a stiffness of 2.75 x 10^6 in-lb/rad. Figure D-2 shows the interface impedance of the Falconsat spacecraft as the moment produced per g of lateral input acceleration. Note that the apparent moment at 10 Hz is approximately equal to the model weight (25 pounds). The damping factor has been estimated from actual test data to be approximately 0.013. However, a more conservative estimate of 0.026 has been used in the model.

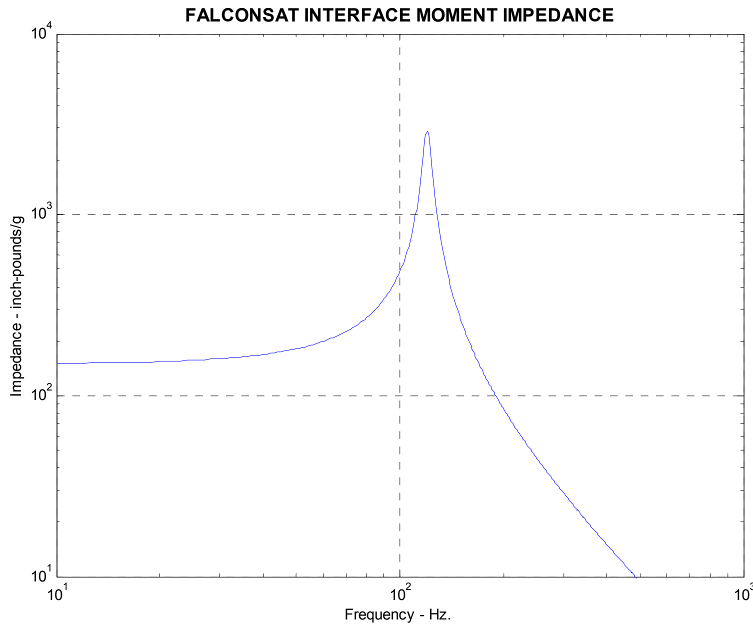


Fig. D-2. Falconsat Interface Moment Impedance.

Figure D-3 shows the relative impedances of the Hitchhiker-PES interface and the Falconsat.

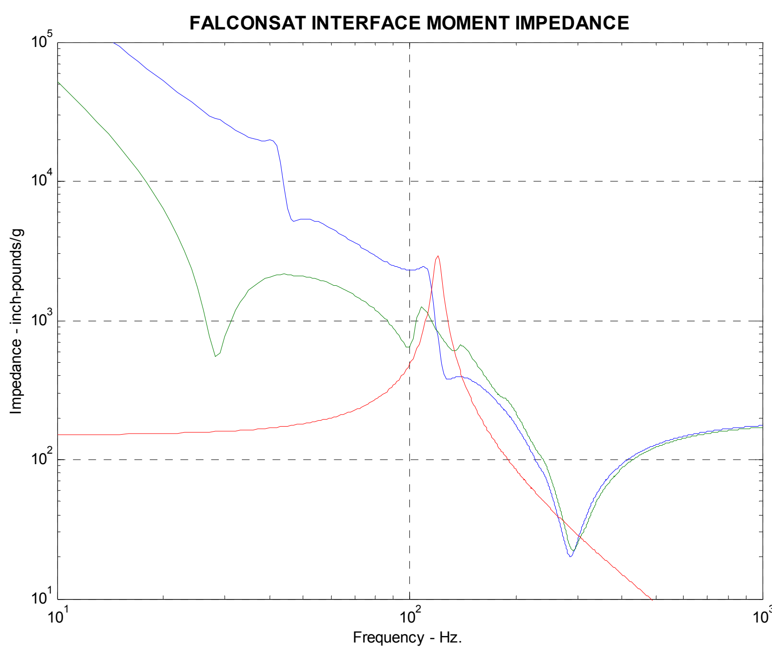


Fig. D-3. Relative Impedances of the Hitchhiker-PES Interface and the Falconsat.

The magnitude of the notch has been computed using equation 3 and the maximum impedance of the PES interface, shown in figure 1, and the impedance of the Falconsat model, shown in figure 3. Figure D-4 shows the magnitude of the notch for the Falconsat model with a 120 Hz. resonance and damping set to 0.026.

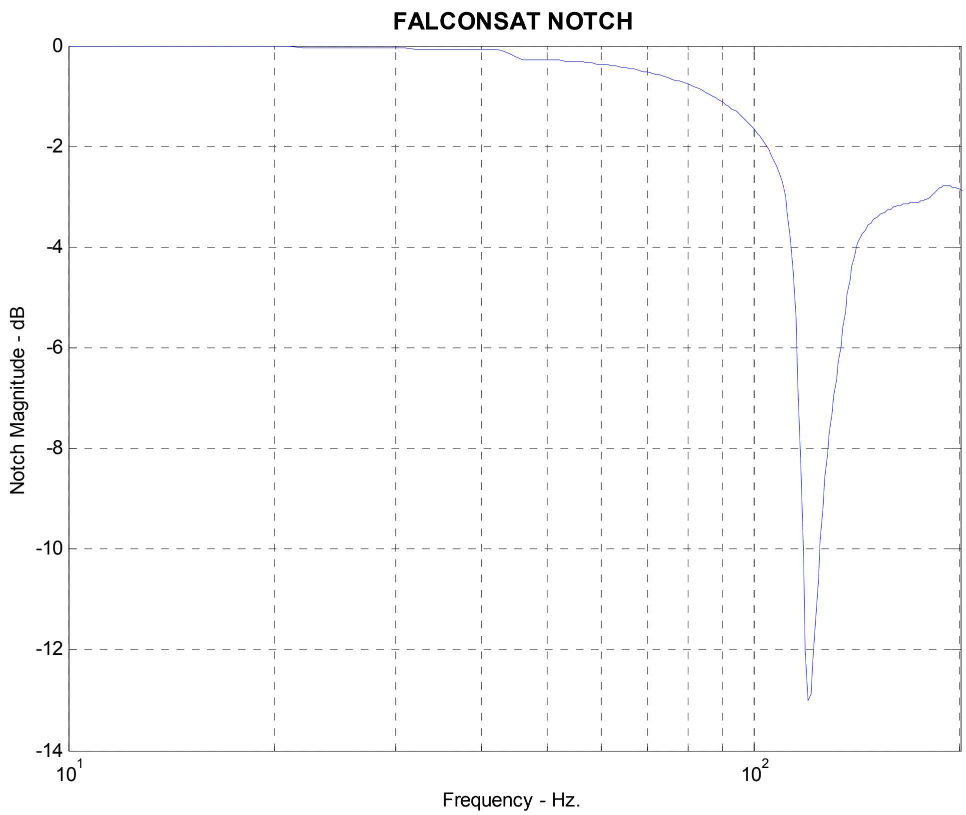


Fig. D-4. Falconsat Notch – Resonant Frequency=120 Hz., zeta=0.026.

Figure D-5 demonstrates the variation of the notch as the resonant frequency of the Falconsat model is varied over the frequency range from 100 to 200 Hz. The resonant frequency was changed by varying the torsional spring stiffness. All other mass properties were held constant. As above, the damping coefficient was set at 0.026. The magnitude of the notch was computed for each model and overlaid on this plot. The plot shows that the magnitude of the notch is approximately about 12 dB at 120 Hz. and increases as the resonant frequency is increased.

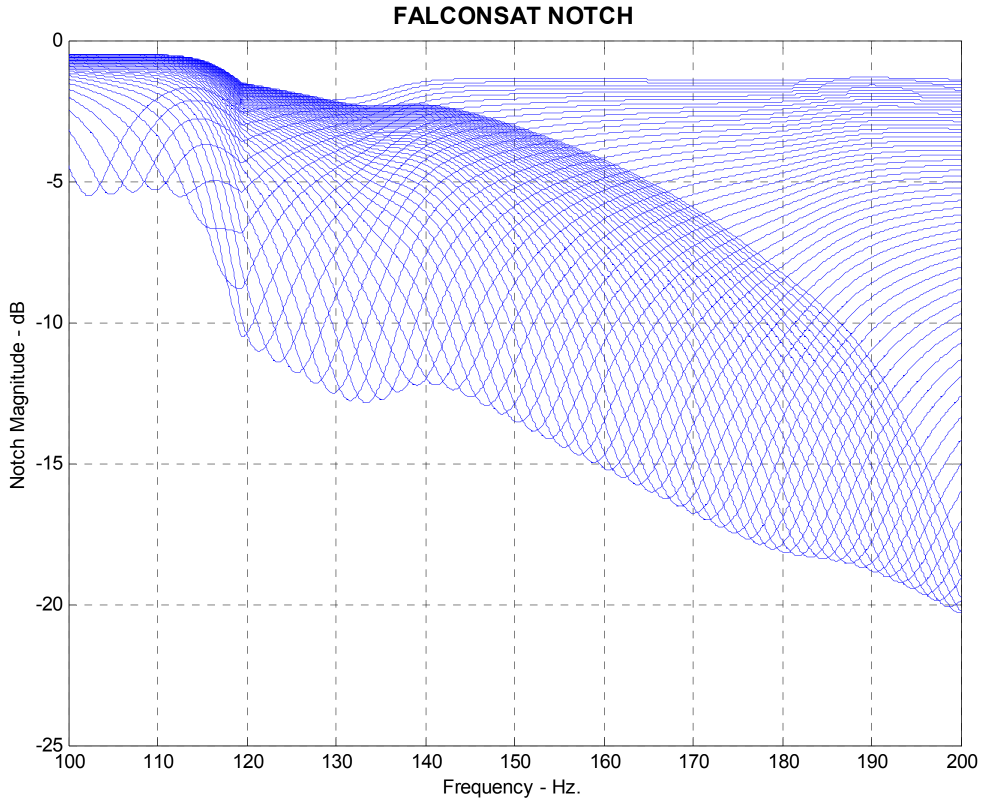


Fig. 5. Notch Envelope – Falconsat Resonant Frequency 100-200 Hz., zeta=0.026.

FalconSat-2

Structural Verification Report

Appendix E

Key Data from the Qualification Vibration Test

Tom Sarafin

May 6, 2002

Contents:

FalconSat-2 SVR, Appendix E, Key Data from the Qualification Vibration Test

E.1 Data Plots

Figure E-1 shows locations of accelerometers used in the qualification test.

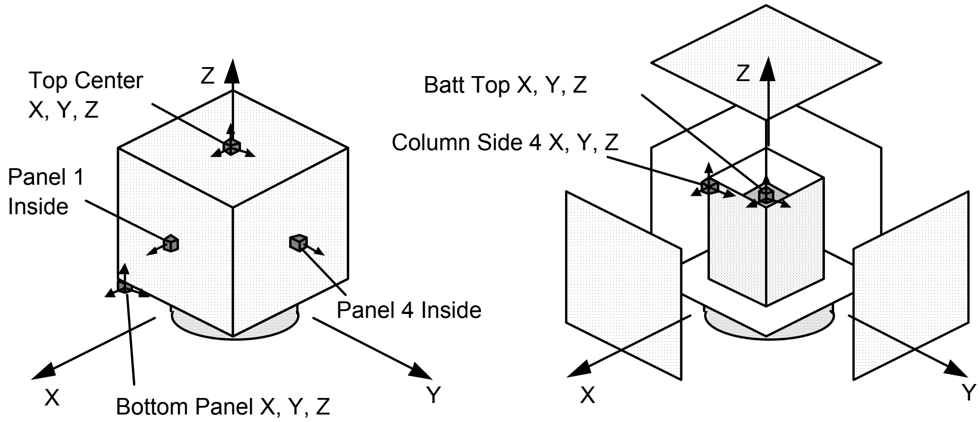


Fig. E-1. Accelerometer Locations.

Figures E-2 through E-18 are key data plots.

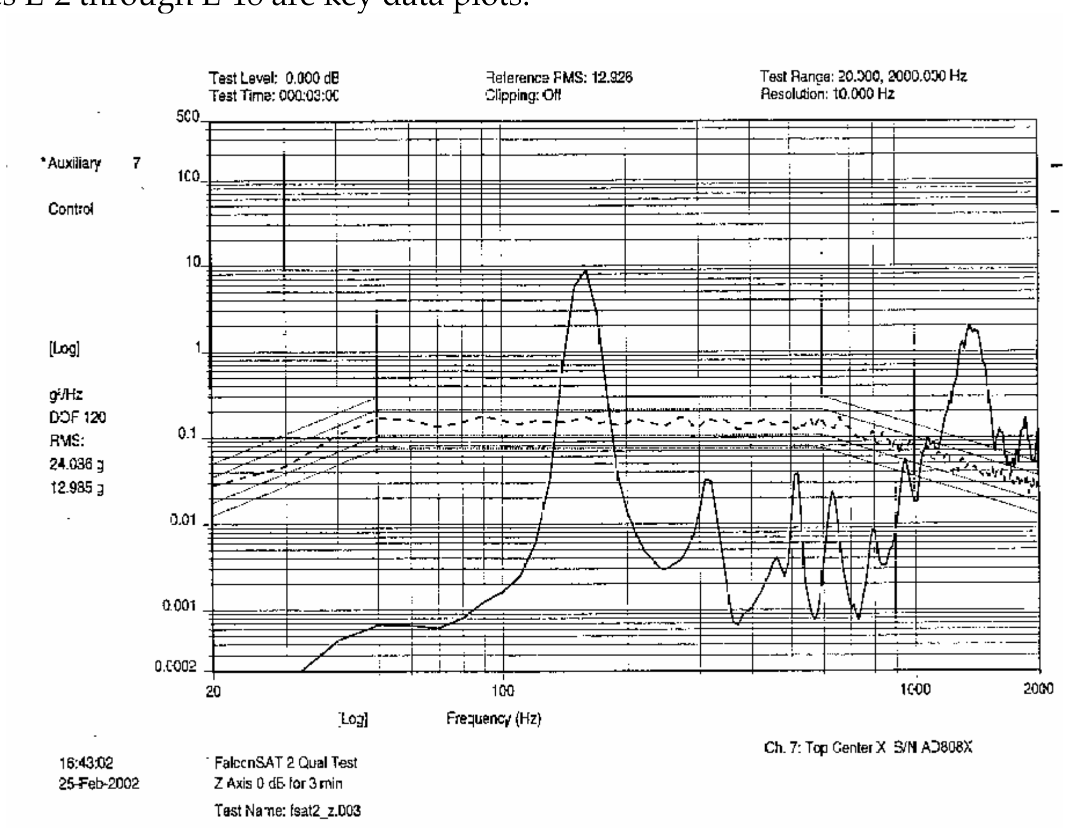


Fig. E-2. Z-axis Random Vibration, Full Qualification Level, Top Center X.

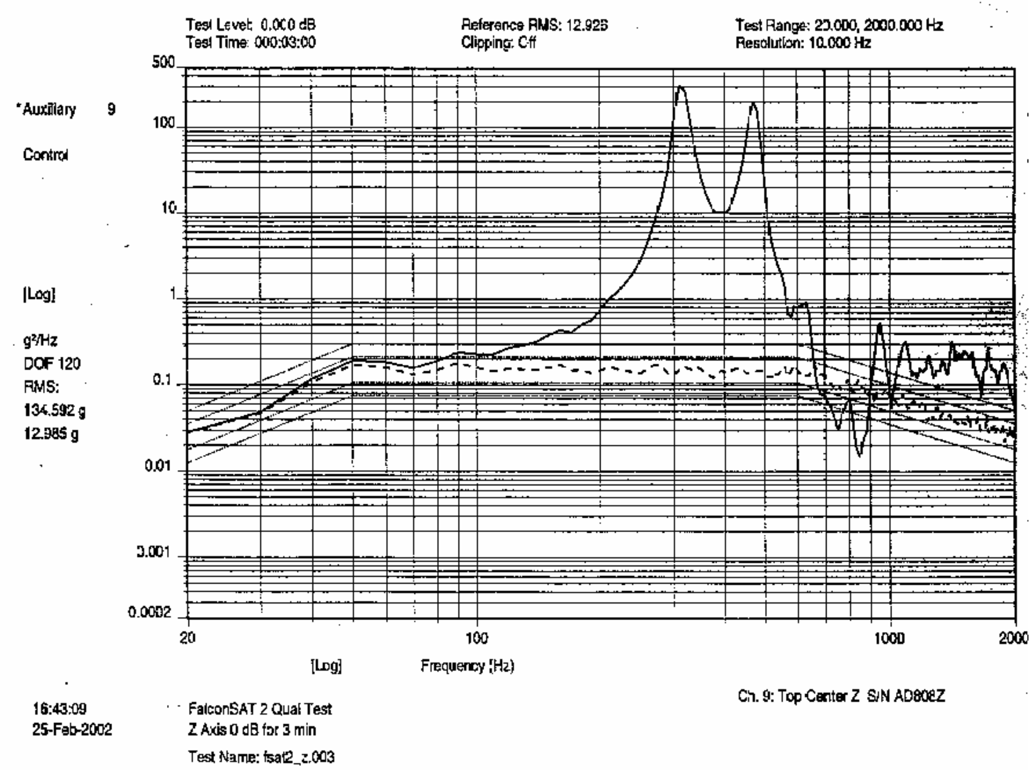


Fig. E-3. **Z-axis Random Vibration, Full Qualification Level, Top Center Z.**

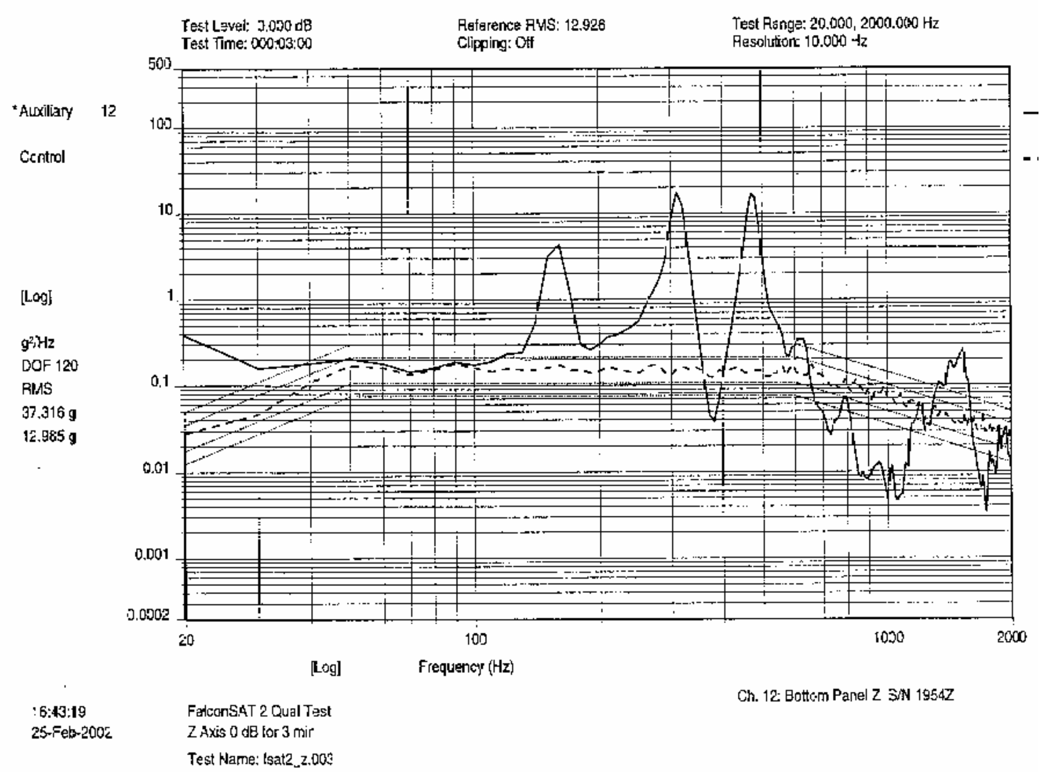


Fig. E-4. **Z-axis Random Vibration, Full Qualification Level, Bottom Corner Z.**

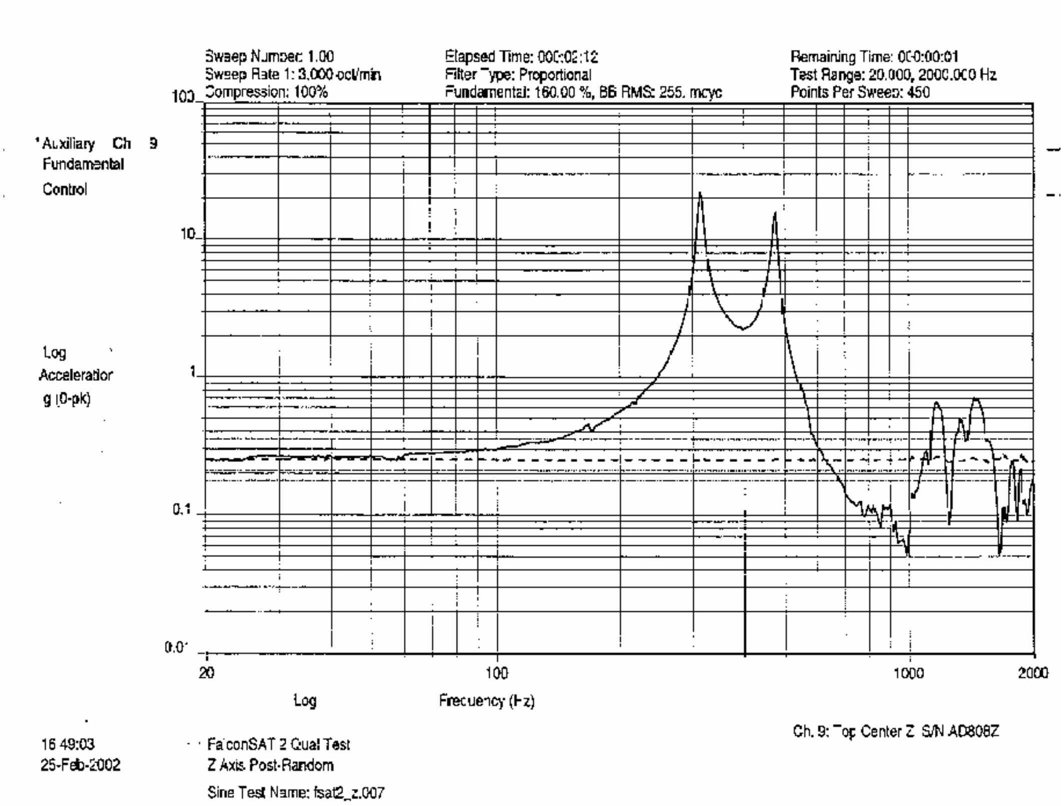


Fig. E-5. Z-axis Post-Test Sine Sweep, Top Center Z.

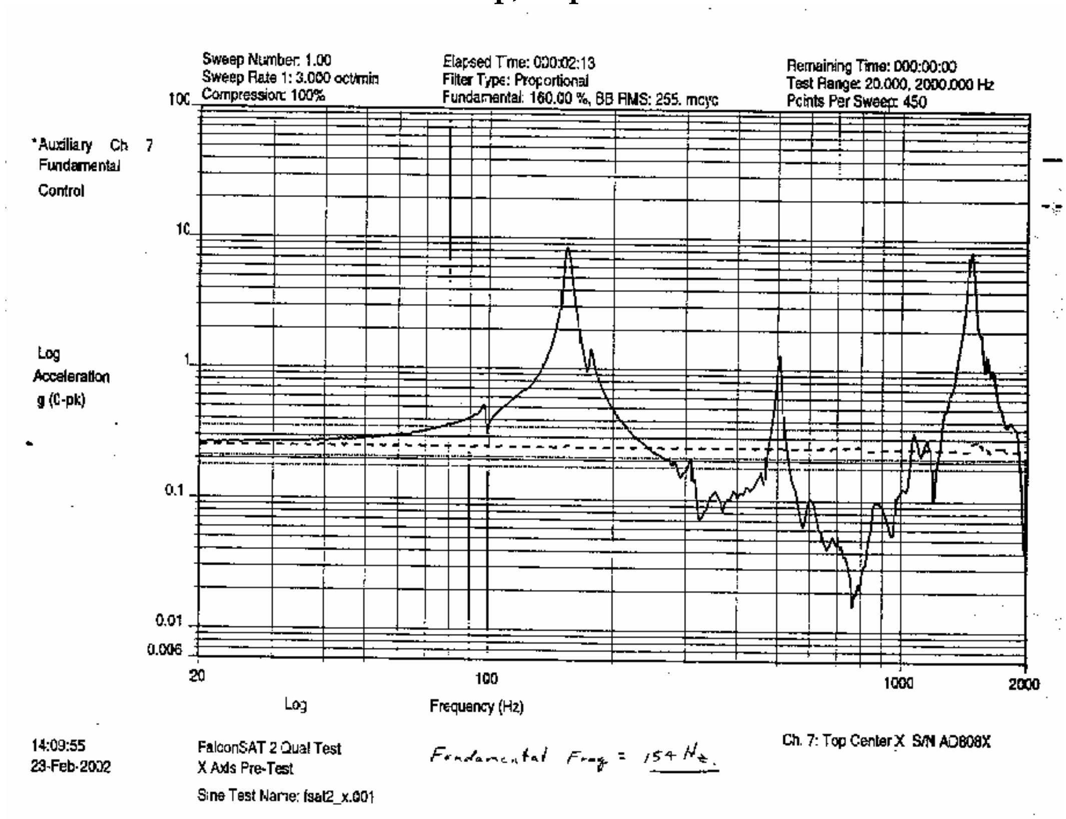


Fig. E-6. X-axis Pretest Sine Sweep, Top Center X.

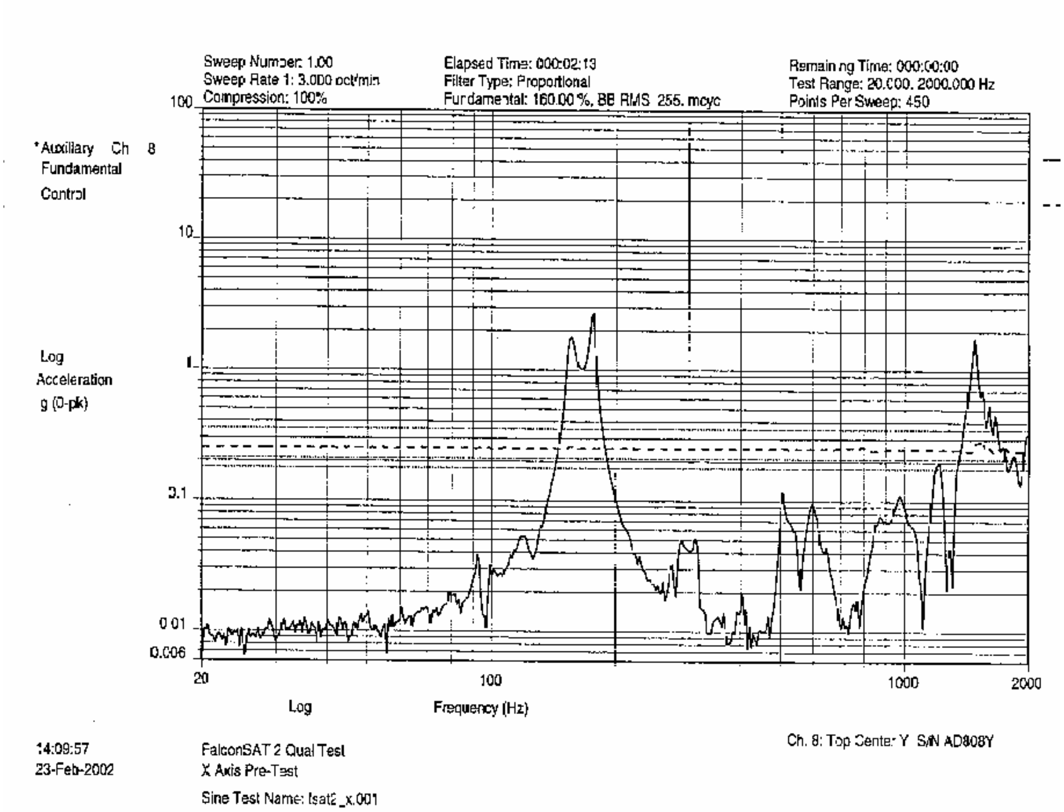


Fig. E-7. **X-axis Pretest Sine Sweep, Top Center Y.**

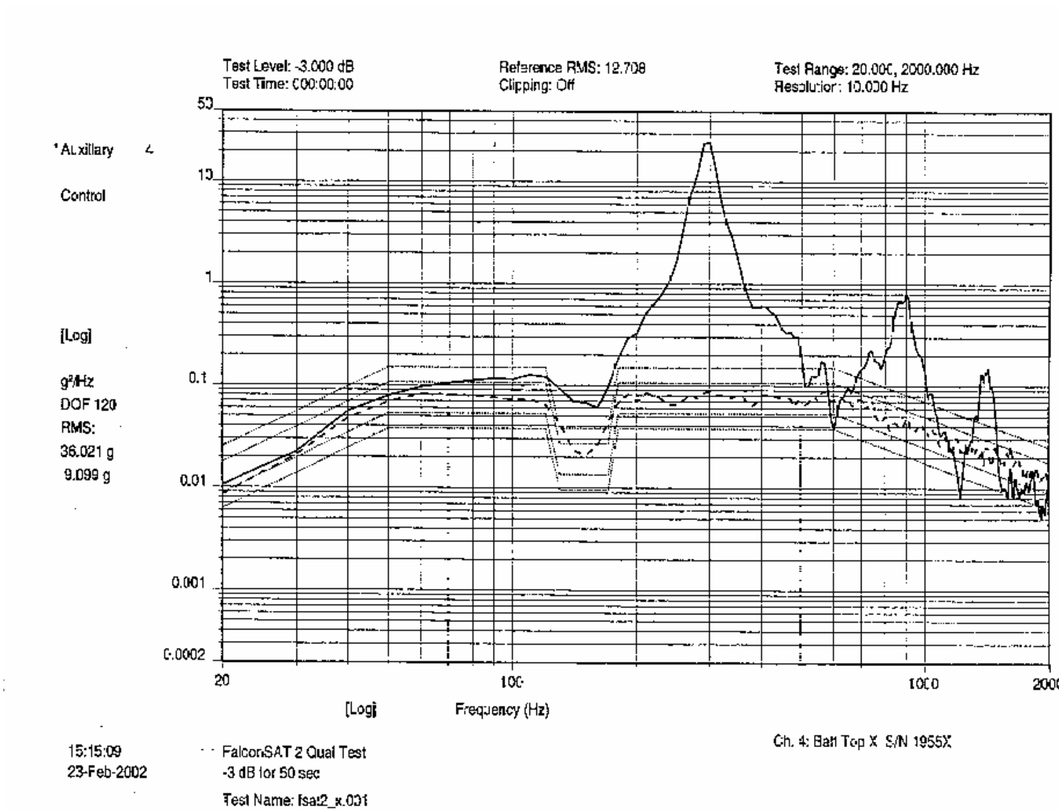


Fig. E-8. **X-axis Random Vibration, Acceptance Level, Battery Top X.**

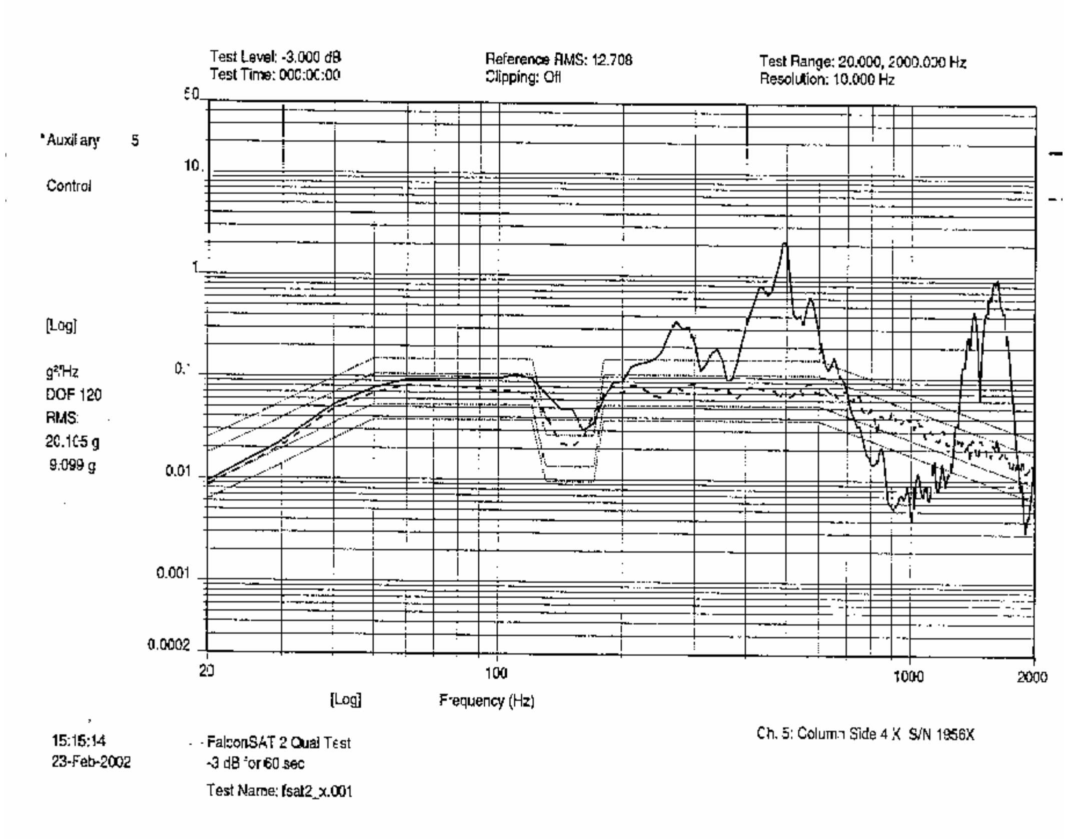


Fig. E-9. X-axis Random Vibration, Acceptance Level, Column Top X.

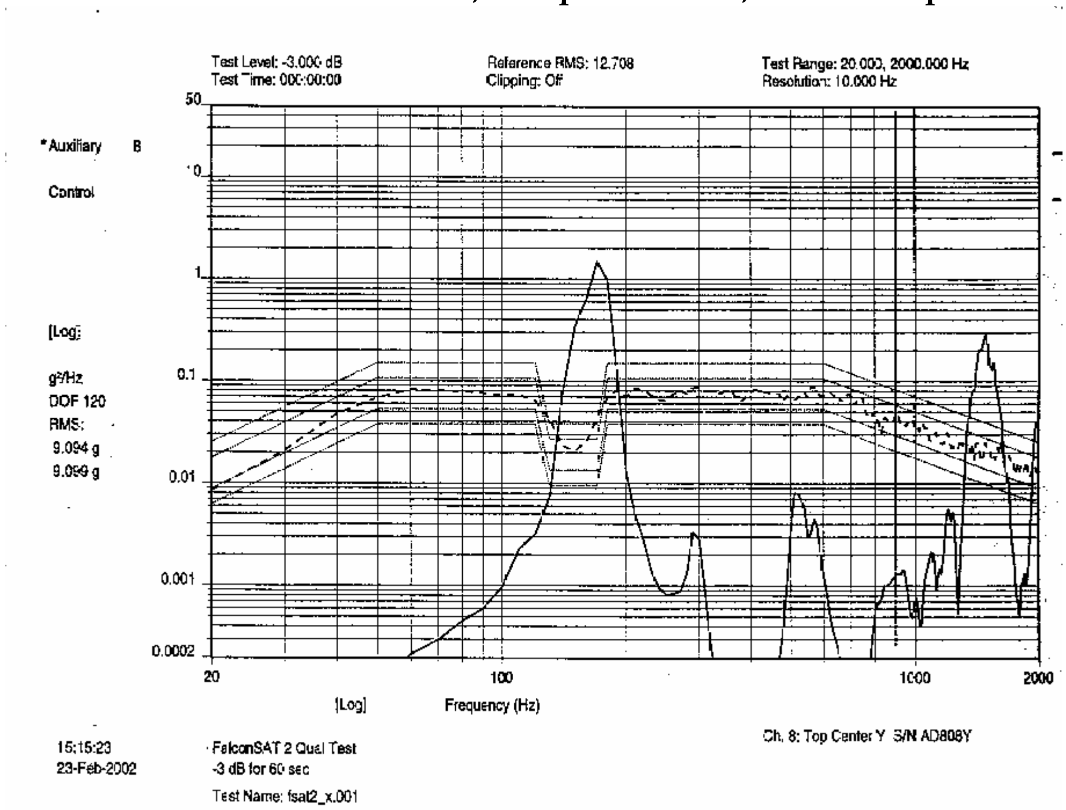


Fig. E-10. X-axis Random Vibration, Acceptance Level, Top Center Y.

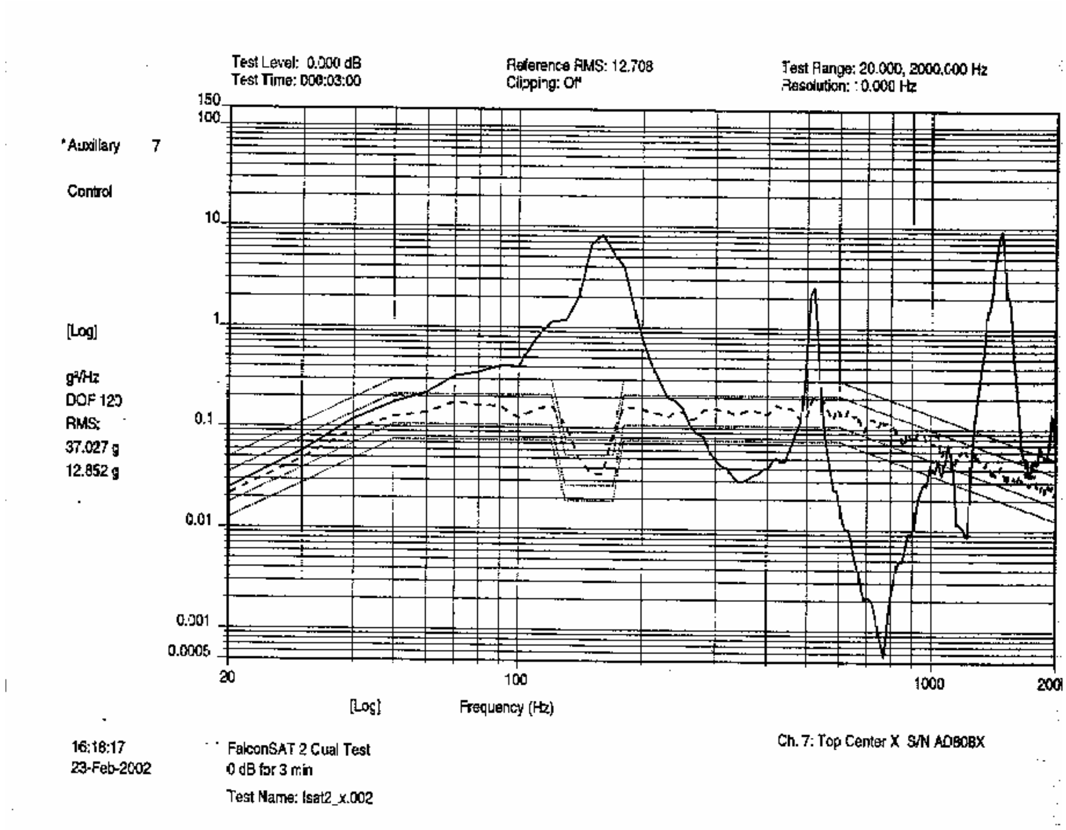


Fig. E-11. X-axis Random Vibration, Full Qualification Level, Top Center X.

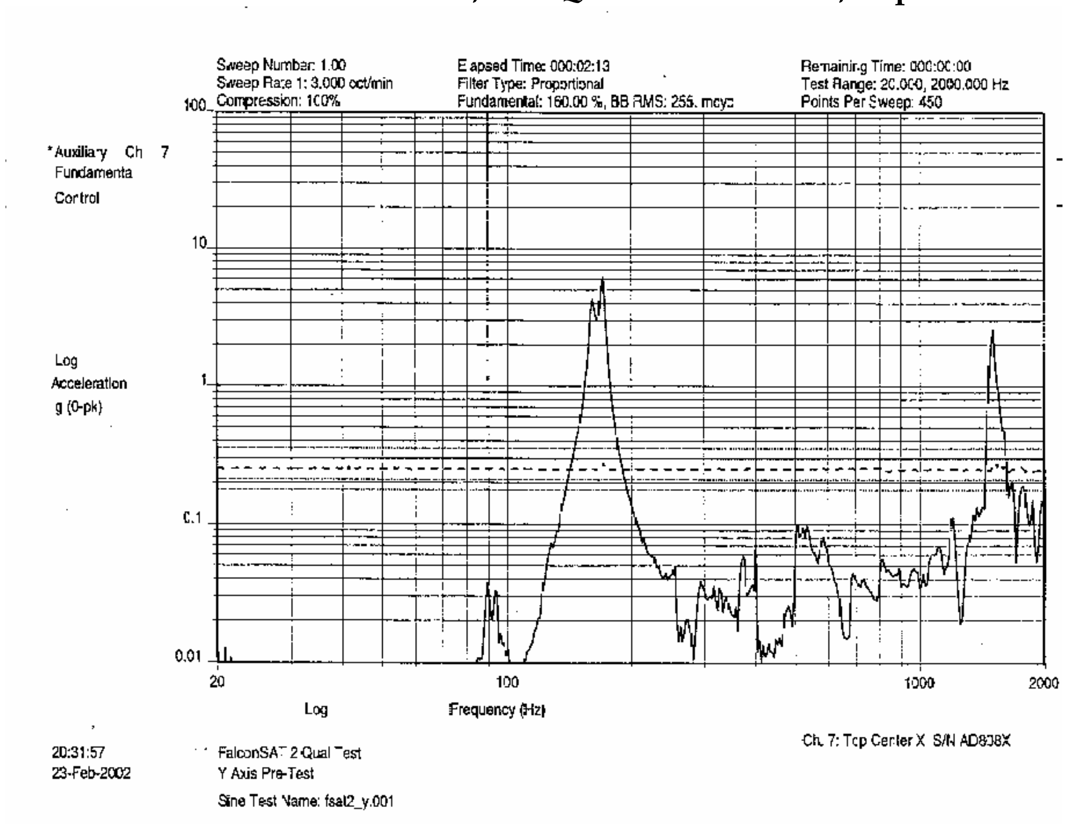


Fig. E-12. Y-axis Pretest Sine Sweep, Top Center X.

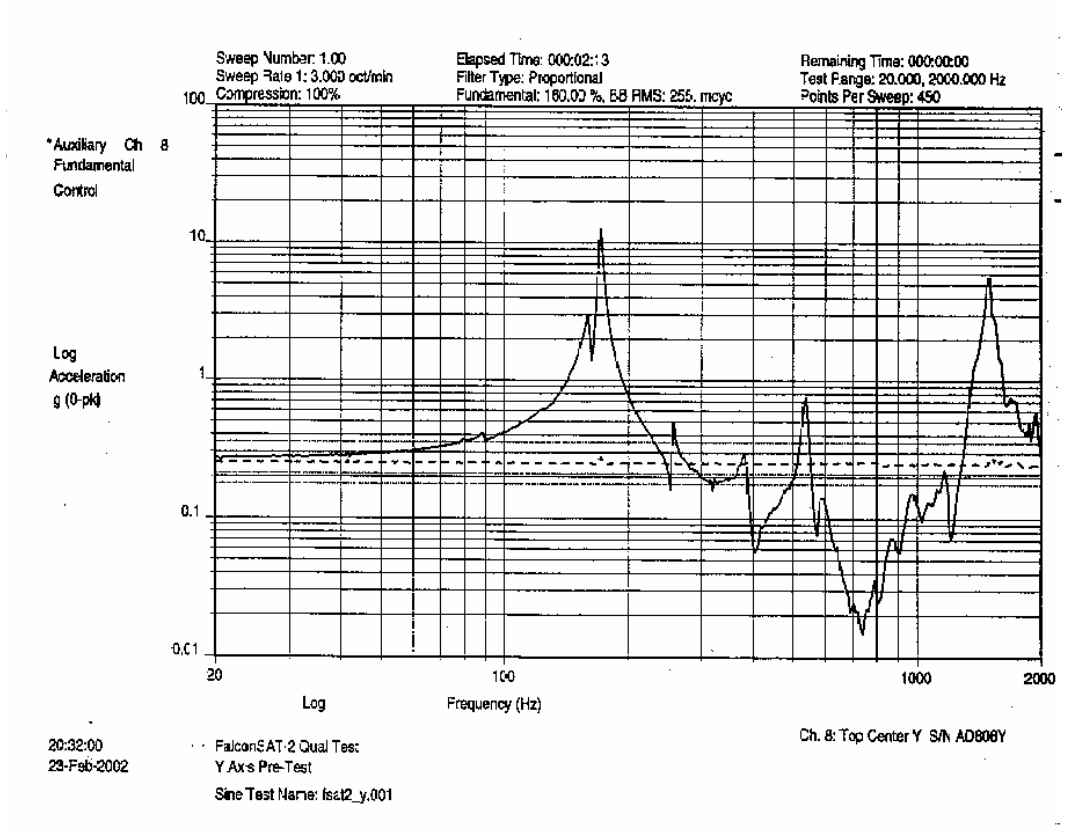


Fig. E-13. Y-axis Pretest Sine Sweep, Top Center Y.

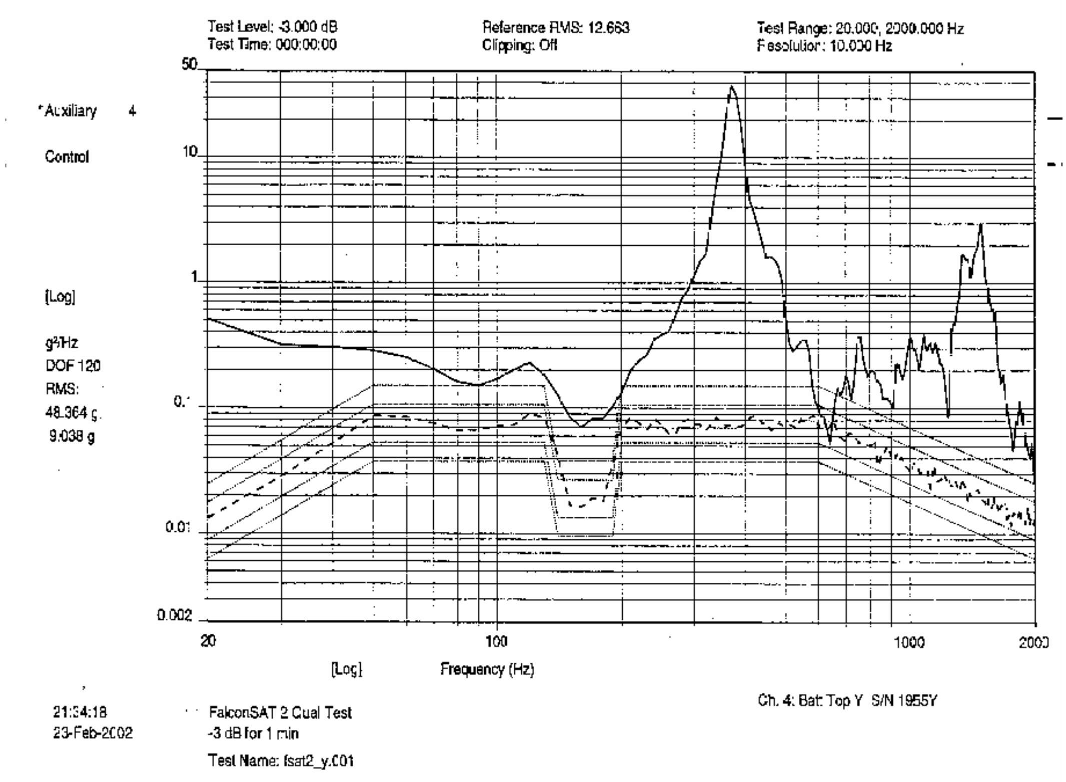


Fig. E-14. Y-axis Random Vibration, Acceptance Level, Battery Top Y.

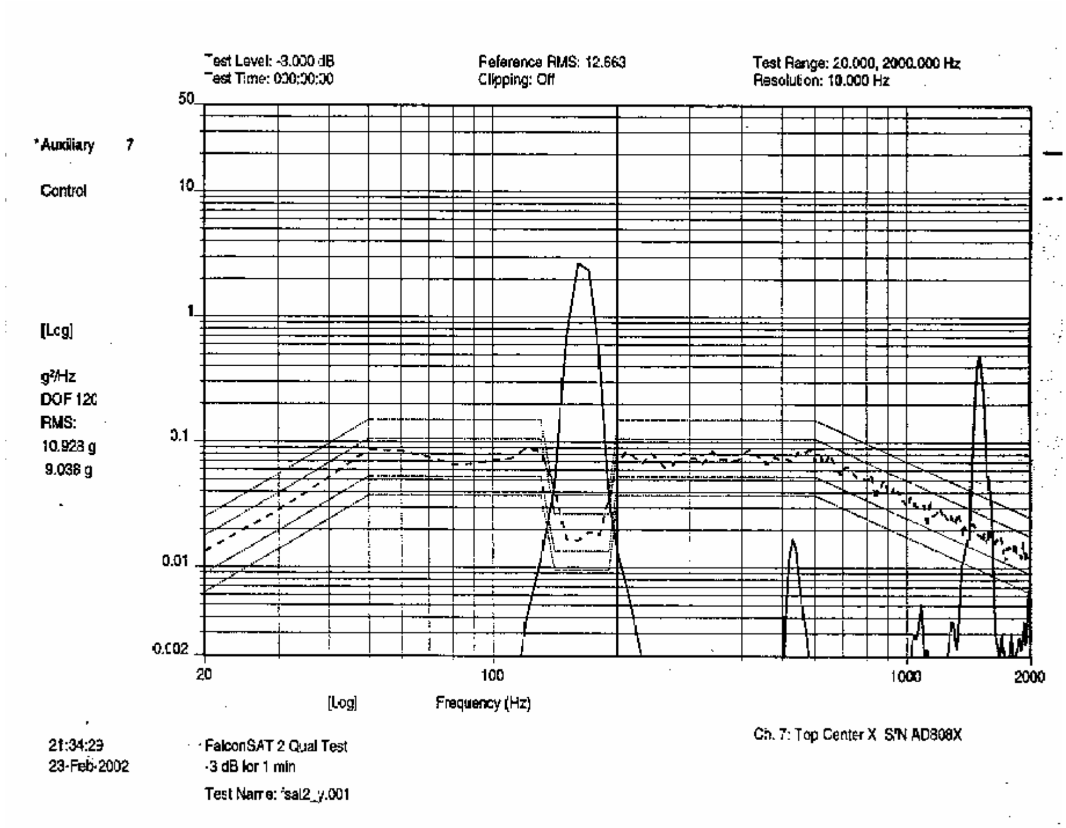


Fig. E-15. Y-axis Random Vibration, Acceptance Level, Top Center X.

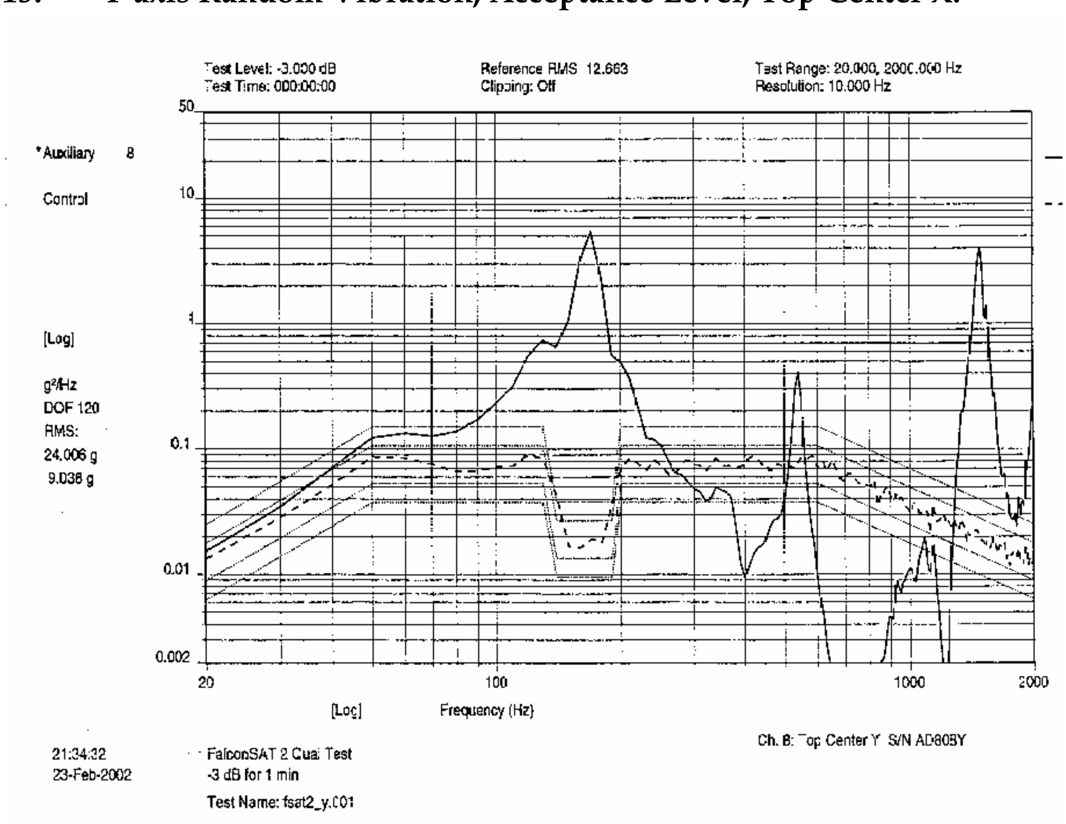


Fig. E-16. Y-axis Random Vibration, Acceptance Level, Top Center Y.

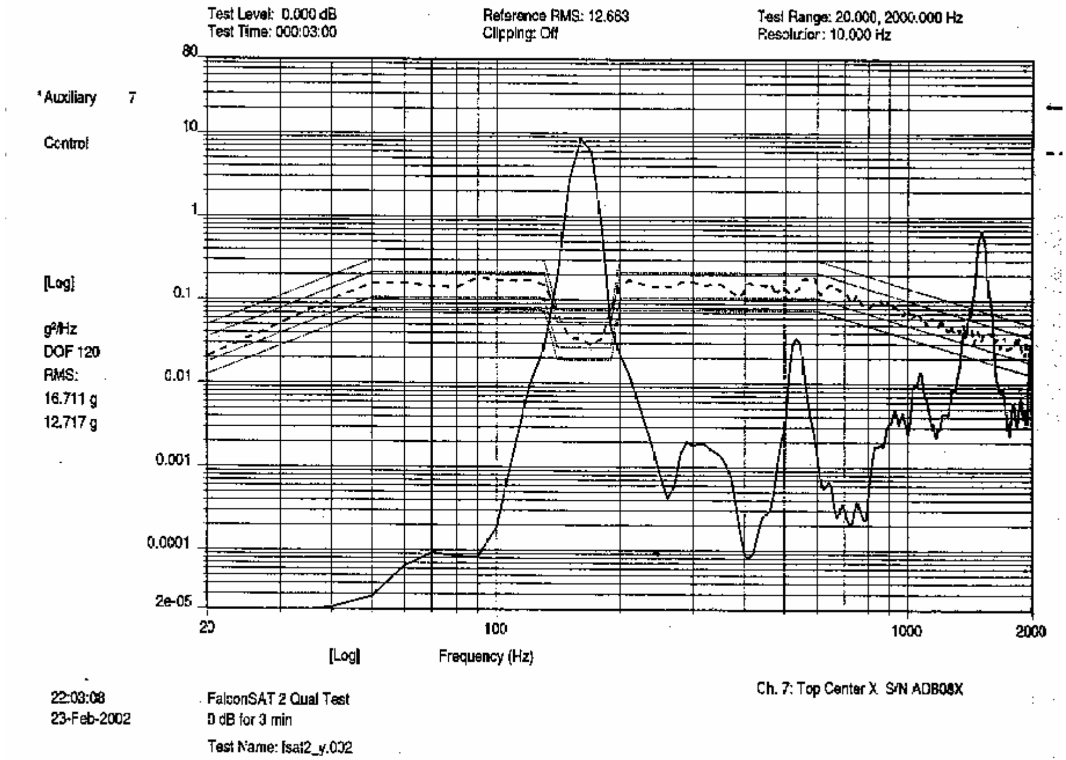


Fig. E-17. Y-axis Random Vibration, Full Qualification Level, Top Center X.

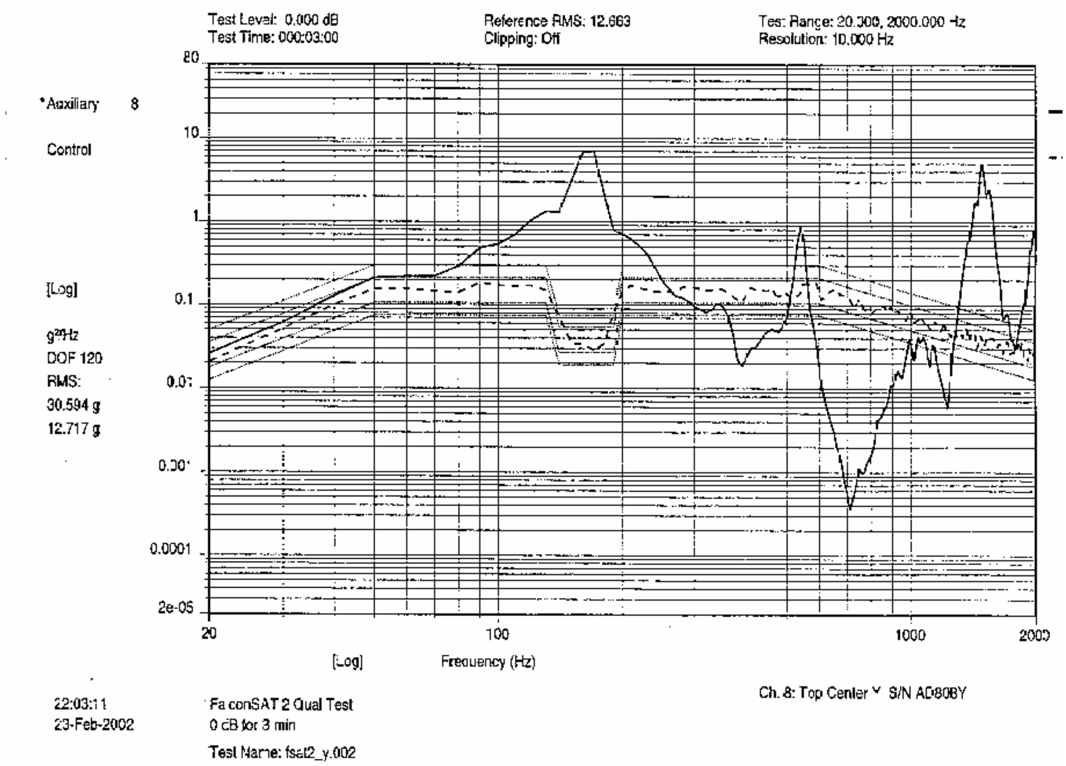


Fig. E-18. Y-axis Random Vibration, Full Qualification Level, Top Center Y.

E.2 Loads Derivation

This section derives loads for use in Appendix A. Spreadsheets are used to numerically integrate applicable portions of selected response power spectral densities (PSDs) to determine the root-mean-square (RMS) acceleration of interest. The RMS acceleration is the square root of the area under the PSD.

E.2a Derivation of RMS Z Acceleration of the Exterior Box Portion of FS2 for Qualification-Level Z-Axis Random Vibration.

Numerical integration is not necessary to estimate the RMS acceleration. Fig. E.4 shows the full RMS level is 37.316 g. From the response PSD, it appears most of this RMS acceleration is from the two highest peaks, at 310 Hz and 480 Hz, which correspond to the first two axial modes for FS2 (as indicated by the finite-element model—see Appendix C). The first hump in the response PSD is associate with rocking from the fundamental X and Y modes, which are excited at relatively low levels by Z input. Discounting this hump and the high-frequency response of the test fixture (jagged peaks at around 1500 Hz), I will assume the qual-level RMS axial acceleration of the exterior box portion to be 35 g.

E.2b Derivation of RMS X Acceleration at the Top of FS2 for Qualification-Level X-Axis Random Vibration.

Of interest is the response of the 157-Hz fundamental rocking mode because only this mode causes significant stress in the base plate and moment at the separation-ring interface. Figure E-11 shows the X-axis reponse PSD at the top of the box for full qualification-level excitation in the X axis. The spreadsheet in Table E-1 is a numerical integration of the peak in Fig. E-11 associated with the fundamental rocking mode.

TABLE E-1. RMS X-axis Response at Top of Box for Qualification-Level X-axis Input.

FalconSat-2: random vibe test of qual model, X axis, 2/23/02 Qual level Tom Sarafin
Response of fundamental rocking mode at 154 Hz, center of top panel

This spreadsheet numerically integrates a response PSD and calculates the rms response as the square root of the area under the curve

Frequency Range (Hz)		Midpoint freq (Hz)	g2/Hz	bandwidth (Hz)	Area (g2)
20	50	35	0.09	30	2.7
50	100	75	0.33	50	16.5
100	118	109	0.8	18	14.4
118	130	124	1.15	12	13.8
130	140	135	3.5	10	35
140	160	150	7.5	20	150
160	175	167.5	5.5	15	82.5
175	200	187.5	2.5	25	62.5
200	225	212.5	0.4	25	10
225	250	237.5	0.15	25	3.75
		Sum			391.15

rms from first rocking mode: 19.7775125 (from 20 to 250 Hz)

E.2c Derivation of RMS Y Acceleration at the Top of FS2 for Qualification-Level Y-Axis Random Vibration.

Of interest is the response of the 171-Hz fundamental rocking mode because only this mode causes significant stress in the base plate and moment at the separation-ring interface. Figure E-18 shows the X-axis reponse PSD at the top of the box for full qualification-level excitation in the X axis. The spreadsheet in Table E-2 is a numerical integration of the peak in Fig. E-18 associated with the fundamental rocking mode.

TABLE E-2. RMS Y-axis Response at Top of Box for Qualification-Level Y-axis Input.

FalconSat-2: random vibe test of qual model, Y axis, 2/23/02 Qual level Tom Sarafin
Response of fundamental rocking mode at 171 Hz, center of top panel

This spreadsheet numerically integrates a response PSD and calculates the rms response as the square root of the area under the curve

Frequency Range (Hz)		Midpoint freq (Hz)	g2/Hz	bandwidth (Hz)	Area (g2)
20	50	35	0.09	30	2.7
50	100	75	0.3	50	15
100	118	109	0.7	18	12.6
118	130	124	1.1	12	13.2
130	140	135	1.5	10	15
140	160	150	4	20	80
160	175	167.5	7	15	105
175	200	187.5	2.5	25	62.5
200	225	212.5	0.6	25	15
225	250	237.5	0.3	25	7.5
		Sum			328.5
		rms from first rocking mode:			18.124569 (from 20 to 250 Hz)

E.2d Derivation of RMS Lateral (X or Y) Acceleration at the Top of the Internal Column for Qualification-Level X- or Y-Axis Random Vibration.

Figure E-9 shows the total RMS X-axis acceleration at the top of the column in response to qualification X-axis input is 20.1 g. A conservative assumption is that this acceleration is all associated with the fundamental rocking mode of the column. Because this acceleration is relatively low, greater accuracy is not necessary, so numerical integration will not be done.

E.2e Derivation of RMS X Acceleration at the Top of the Battery for Qualification-Level X-Axis Random Vibration.

Figure E-8 shows the fundamental rocking frequency of the battery is 300 Hz. Table E-3 is a numerical integration of the response PSD associated with the large peak at 300 Hz. This integration is for acceptance-level excitation; at qualification levels, this accelerometer malfunctioned. Assume the qualification-level RMS response is 1.4 times the acceptance-level value. Thus, qual-RMS = 1.4(31.4) = 44 g.

TABLE E-3. RMS X-axis Response at Top of Battery for Acceptance-Level X-axis Input.

FalconSat-2: random vibe test of qual model, X axis, 2/23/02 — Acceptance level — Tom Sarafin
Response of battery mode at 300 Hz, acceleration measured at battery top

This spreadsheet numerically integrates a response PSD and calculates the rms response as the square root of the area under the curve

Frequency Range (Hz)		Midpoint freq (Hz)	g2/Hz	bandwidth (Hz)	Area (g2)
160	200	180	0.2	40	8
200	230	215	0.6	30	18
230	260	245	2	30	60
260	290	275	10	30	300
290	300	295	23	10	230
300	320	310	10	20	200
320	350	335	3	30	90
350	380	365	0.9	30	27
380	420	400	0.6	40	24
420	500	460	0.4	80	32
		Sum			989
		rms from first rocking mode:			31.4483704 (from 160 to 500 Hz)

E.2f Derivation of RMS Y Acceleration at the Top of the Battery for Qualification-Level Y-Axis Random Vibration.

Figure E-14 shows the fundamental rocking frequency of the battery is 370 Hz in this axis. Table E-4 is a numerical integration of the response PSD associated with the large peak at 370 Hz. This integration is for acceptance-level excitation. Assume the qualification-level RMS response is 1.4 times the acceptance-level value. Thus, qual-RMS = 1.4(44.2) = 62 g.

TABLE E-4. RMS X-axis Response at Top of Battery for Acceptance-Level X-axis Input.

FalconSat-2: random vibe test of qual model, Y axis, 2/23/02 Acceptance level Tom Sarafin

Response of battery mode at 370 Hz, acceleration measured at battery top

This spreadsheet numerically integrates a response PSD and calculates the rms response as the square root of the area under the curve

Frequency Range (Hz)		Midpoint freq (Hz)	g2/Hz	bandwidth (Hz)	Area (g2)
200	250	225	0.25	50	12.5
250	300	275	0.8	50	40
300	320	310	1.5	20	30
320	340	330	6	20	120
340	360	350	20	20	400
360	380	370	35	20	700
380	400	390	22	20	440
400	440	420	3.5	40	140
440	500	470	1.2	60	72
		Sum			1954.5
		rms from first rocking mode:			44.209727 (from 200 to 500 Hz)

FalconSat-2
Structural Verification Report

Appendix F
Insert Pull-out Test Report

Tom Sarafin

May 16, 2002

Summary

This appendix documents the derivation of a design allowable pull-out strength for Spiralock inserts for #10 screws corresponding to the insert-installation process used at the United States Air Force Academy for the FalconSat program. The resulting allowable, which was derived from test, applies for inserts of size "one diameter long" installed in 6061-T651 aluminum alloy. The allowable was derived to meet the intent of the "A-basis" definition given in MIL-HDBK-5G: 99% probability at 95% statistical confidence.

The test data (see Sec. F.3, which is a Microsoft Excel spreadsheet in a separate electronic file, named "FS2_SVR_Apx_SecF3") also indicates, from load-deflection curves, the load at which the joint gapped; i.e., the applied load at which there is no remaining clamping force between the fittings. This is useful information for ensuring the joints in the flight hardware will not gap at limit load.

Results:

- Allowable ultimate strength is 1900 lb.
- Average bolt preload for 60 in-lb torque is 1220 lb (ref. Sec. F.3)

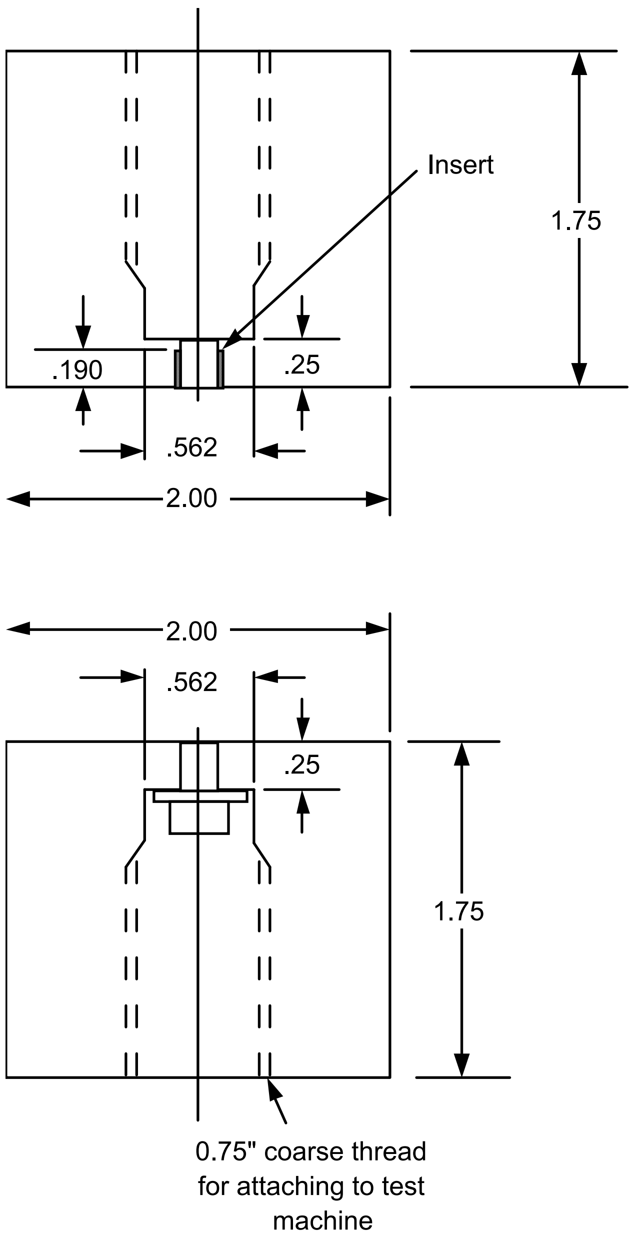
Key conclusion: The test demonstrated that bolt preload does not affect joint strength, so preload need not be included in strength analysis.

Reference: MIL-HDBK-5G—"Metallic Materials and Elements for Aerospace Vehicle Structures." Department of Defense.

F.1 Test Description

The FalconSat-2 (FS2) structure uses Spiralock inserts for #10 (0.190" dia.) bolts in several locations. A test was run on March 19, 2002, at the United States Air Force Academy (USAFA) to determine a design allowable pull-out strength for these inserts.

The parts shown in Fig. F-1 were machined from 6061-T6 extrusion to be consistent with the 6061-T651 plate used for the FS2 structure. The two types of parts were identical except, for the bolt hole in the part depicted by the upper sketch was tapped, and an insert was installed. Twelve of each type of part were fabricated to support twelve tests, which was considered adequate for capturing variation in the process of installing inserts.



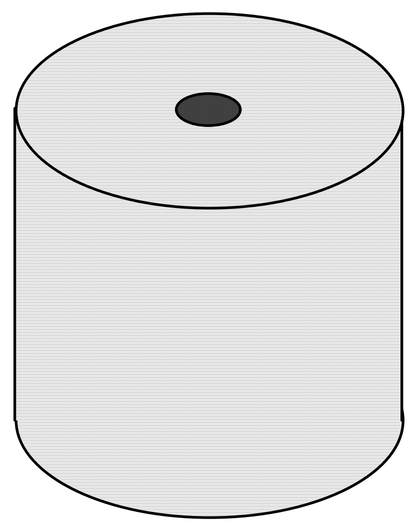


Fig. F-1. Test Specimens, with Bolt Configuration Shown.

The inserts used were Spiralock part no. 96100, same as are used for flight hardware. This is a wire-type insert that provides an internal 10-32 thread for #10 bolts and requires a tapped hole with a major diameter of 0.2306". The insert is referred to as a "one-diameter" insert because its length, or depth of engagement in the parent material, is approximately equal to the diameter of the bolt (0.180" for the insert length vs. 0.190" diameter). The insert material was austenitic stainless steel.

Inserts were installed using the same USAFA process as is used for flight hardware, which is based on the process recommended by the insert manufacturer.

The bolts and washers used in the test were also the same as for flight hardware: fully threaded NAS1351 bolts (160-ksi A-286) and countersunk NAS1587 A3C washers. For reference, the bolt-head diameter was 0.303/0.312", and the washer outside diameter and thickness were 0.471" and 0.064", respectively.

Tests were done with varying installation torque so we could understand the effects of bolt preload on strength under external load. Four specimens had bolts installed snug only, four had bolts torqued to 60 +/-3 in-lb (the torque used for flight hardware), and four had an installation torque of 100 +/-3 in-lb. As a point of reference, earlier tests at USAFA showed that these bolts failed during installation in these inserts when torqued to approximately 120 in-lb.

Load was applied by a tensile-test machine at a rate of about 800 lb/sec (in the linear-elastic region). (The machine actually controlled the rate of deflection.)

F.2 Test Results and Conclusions

In all tests, failure was quite ductile, and testing continued until the applied load had dropped dramatically from the peak. The failure mode was thread stripping, although it visually appeared that it was the external threads of the bolt that failed, not the internal threads in the parent material. This was surprising, given the relatively low shear strength of 6061-T6 aluminum alloy.

Table F-1 summarizes the peak loads achieved in each test, and Table F-2 shows key statistical values. We had expected the specimens with the highest bolt preload to have an average strength under applied load either equal to or slightly less than the average strength for the specimens with bolts having no preload. The test data shows otherwise. It makes no sense that a preloaded joint can carry <u>more</u> external tensile load than a nonpreloaded bolt because the bolt load is never less than the applied load in a preloaded joint. However, there was enough plastic deformation associated with the failure that, at peak load, any clamping force was gone, meaning the total load in the bolt was the applied load. This being the case, we concluded that preload has no effect on strength, and the difference in average values shown in Table F-2 is simply a statistical quirk of small sample sizes. Thus, to determine an allowable load for design, we decided to use the full set of data for twelve tests.

A key conclusion from the test is that, because bolt preload does not affect joint strength, preload need not be included in strength analysis.

Table F-1. Test Results.

Test Number	Installation Torque (in-lb)	Ultimate Load (lb)
1	0	2481
2	0	2495
3	0	2419
4	0	2318
5	60	2635
6	60	2397
7	60	2376
8	60	2410
9	100	2633
10	100	2552
11	100	2524
12	100	2534

Table F-2. Statistical Values from Test Results.

Test Groups	Sample Mean Load (lb)	Unbiased Sample Standard Deviation (lb)
1 - 4	2428	81
5 - 8	2455	121
9 - 12	2561	50
All tests	2481	100

According to Table 9.6.4.1 of MIL-HDBK-5G, the one-sided tolerance limit factor for the normal distribution is 3.747 when the sample size is 12. The A-basis allowable is defined as

$$P = \mu - 3.747\sigma$$

where μ is the sample mean and σ is the unbiased sample standard deviation. Because the test specimens were not truly statistically independent, all using the same lot of parent material, I will reduce this value by 10% to be on the safe side. Thus, the allowable ultimate pull-out strength is

$$P = 0.9[2481 - 3.747(100)] = 1900 \text{ lb}$$

FalconSat-2 Structural Verification Report, Appendix F, Section F.3
Bolt/Insert Pull-out Test, USAFA--Reduction of Test Data 3/19/2002

This spreadsheet includes the raw test data (included under the columns titled "Displ." for displacement and "Load") and reduced data.
- "Normalized Displ." is equal to the measured displacement at a load increment minus the measured displacement at the initial load increment.
- "Estimated Joint Displ." is an estimated displacement of the joint locally, equal to the total displacement minus the estimated displacement of the test equipment.
- "Slope: Load/Displ" is the change in load divided by the change in measured displacement.

Bolt dia:	0.19	inch
Ftu:	160	ksi
Depth:	0.19	inch
Mat'l:	6061-T6	extruded bar, 2" dia.

Load (lb) versus Displacement (in.)

Test # 1 — Torque: Snug

Displ.	Normalized Displ.	Estim. Joint Displ.	Load	Slope: Load/ Displ
0.0105	0.0000	0.00000	102	
0.0133	0.0028	0.00190	201	35500
0.0147	0.0042	0.00258	282	57571
0.0163	0.0058	0.00326	384	63438
0.0179	0.0074	0.00377	505	76000
0.0187	0.0082	0.00397	571	82500
0.0204	0.0099	0.00436	717	86000
0.0212	0.0107	0.00447	794	95875
0.0220	0.0115	0.00453	875	101750
0.0228	0.0123	0.00456	961	106875
0.0245	0.0140	0.00476	1128	98294
0.0253	0.0148	0.00477	1216	109750
0.0261	0.0156	0.00480	1301	106500
0.0270	0.0165	0.00490	1389	97889
0.0278	0.0173	0.00489	1479	112250
0.0286	0.0181	0.00492	1565	107500
0.0303	0.0198	0.00508	1736	100706
0.0311	0.0206	0.00517	1815	98250
0.0320	0.0215	0.00539	1891	84333
0.0336	0.0231	0.00574	2028	86125
0.0345	0.0240	0.00607	2092	70889
0.0361	0.0256	0.00661	2210	73688
0.0378	0.0273	0.00738	2313	60529
0.0395	0.0290	0.00835	2394	47647
0.0411	0.0306	0.00943	2452	36250
0.0428	0.0323	0.01087	2481	17059
0.0461	0.0356	0.01481	2409	-21818
0.0486	0.0381	0.01920	2200	-83600
0.0503	0.0398	0.02311	1955	-144118
0.0520	0.0415	0.02795	1606	-205294

Test # 2 — Torque: Snug

Displ.	Normalized Disp.	Estim. Joint Displ.	Load	Slope: Load/ Displ
0.0127	0.0000	0.00000	101	
0.0199	0.0072	0.00628	201	13889
0.0225	0.0098	0.00798	299	37692
0.0242	0.0115	0.00875	400	59412
0.0256	0.0129	0.00923	500	71429
0.0268	0.0141	0.00951	600	83333
0.0280	0.0153	0.00977	702	85000
0.0292	0.0165	0.01005	802	83333
0.0302	0.0175	0.01015	900	98000
0.0312	0.0185	0.01023	1000	100000
0.0322	0.0195	0.01033	1098	98000
0.0332	0.0205	0.01038	1202	104000
0.0341	0.0214	0.01035	1303	112222
0.0351	0.0224	0.01044	1402	99000
0.0361	0.0234	0.01054	1500	98000
0.0372	0.0245	0.01072	1599	90000
0.0383	0.0256	0.01088	1702	93636
0.0395	0.0268	0.01115	1803	84167
0.0406	0.0279	0.01134	1902	90000
0.0418	0.0291	0.01164	2000	81667
0.0431	0.0304	0.01202	2100	76923
0.0446	0.0319	0.01260	2200	66667
0.0462	0.0335	0.01328	2300	62500
0.0482	0.0355	0.01436	2400	50000
0.0495	0.0368	0.01520	2450	38462
0.0520	0.0393	0.01729	2495	18000
0.0553	0.0426	0.02145	2401	-28485
0.0578	0.0451	0.02578	2202	-79600
0.0594	0.0467	0.02922	2002	-125000
0.0612	0.0485	0.03564	1500	-278889

Test # 3 — Torque: Snug

Displ	Normalized Displ.	Estim. Joint Displ.	Load	Slope: Load/ Displ
0.0104	0.0000	0.00000	102	
0.0133	0.0029	0.00200	201	34138
0.0150	0.0046	0.00276	303	60000
0.0164	0.0060	0.00328	400	69286
0.0177	0.0073	0.00363	503	79231
0.0188	0.0084	0.00384	601	89091
0.0199	0.0095	0.00401	703	92727
0.0209	0.0105	0.00411	801	98000
0.0219	0.0115	0.00419	902	101000
0.0229	0.0125	0.00428	1001	99000
0.0238	0.0134	0.00428	1100	110000
0.0248	0.0144	0.00436	1200	100000
0.0258	0.0154	0.00442	1303	103000
0.0267	0.0163	0.00442	1402	110000
0.0277	0.0173	0.00453	1499	97000
0.0287	0.0183	0.00462	1599	100000
0.0297	0.0193	0.00469	1700	101000
0.0309	0.0205	0.00496	1802	85000
0.0320	0.0216	0.00517	1900	89091
0.0333	0.0229	0.00554	2001	77692
0.0347	0.0243	0.00604	2100	70714
0.0362	0.0258	0.00662	2200	66667
0.0379	0.0275	0.00741	2300	58824
0.0390	0.0286	0.00805	2350	45455
0.0406	0.0302	0.00920	2400	31250
0.0424	0.0320	0.01082	2419	10556
0.0455	0.0351	0.01500	2301	-38065
0.0480	0.0376	0.02022	2004	-118800

Test # 4 — Torque: Snug

Displ	Normalized Disp.	Estim. Joint Displ.	Load	Slope: Load/ Displ
0.0070	0.0000	0.00000	101	
0.0110	0.0040	0.00312	198	24250
0.0133	0.0063	0.00449	300	44348
0.0148	0.0078	0.00506	402	68000
0.0160	0.0090	0.00541	496	78333
0.0171	0.0101	0.00555	601	95455
0.0181	0.0111	0.00570	695	94000
0.0193	0.0123	0.00587	808	94167
0.0203	0.0133	0.00599	904	96000
0.0213	0.0143	0.00608	1004	100000
0.0223	0.0153	0.00614	1108	104000
0.0231	0.0161	0.00611	1199	113750
0.0240	0.0170	0.00616	1292	103333
0.0250	0.0180	0.00617	1401	109000
0.0260	0.0190	0.00622	1506	105000
0.0268	0.0198	0.00623	1592	107500
0.0280	0.0210	0.00639	1707	95833
0.0290	0.0220	0.00655	1799	92000
0.0301	0.0231	0.00672	1901	92727
0.0313	0.0243	0.00705	1997	80000
0.0328	0.0258	0.00759	2102	70000
0.0345	0.0275	0.00842	2198	56471
0.0373	0.0303	0.01027	2302	37143
0.0388	0.0318	0.01163	2318	10667
0.0401	0.0331	0.01310	2299	-14615
0.0425	0.0355	0.01643	2197	-42500
0.0450	0.0380	0.02079	1992	-82000
0.0476	0.0406	0.02806	1479	-197308
0.0510	0.0440	0.03582	1000	-140882
0.0550	0.0480	0.03889	1102	25500

Test # 5 — Torque: 60

Displ	Normalized Disp.	Estim. Joint Displ.	Load	Slope: Load/ Displ
0.0081	0.0000	0.00000	104	
0.0097	0.0016	0.00101	203	61875
0.0109	0.0028	0.00158	307	86667
0.0116	0.0035	0.00173	400	132857
0.0122	0.0041	0.00175	507	178333
0.0127	0.0046	0.00177	589	164000
0.0134	0.0053	0.00179	699	157143
0.0141	0.0060	0.00182	804	150000
0.0147	0.0066	0.00184	898	156667
0.0156	0.0075	0.00207	1010	124444
0.0162	0.0081	0.00213	1099	148333
0.0169	0.0088	0.00229	1190	130000
0.0177	0.0096	0.00243	1300	137500
0.0186	0.0105	0.00271	1404	115556
0.0194	0.0113	0.00291	1504	125000
0.0202	0.0121	0.00314	1598	117500
0.0212	0.0131	0.00348	1709	111000
0.0221	0.0140	0.00385	1797	97778
0.0231	0.0150	0.00426	1895	98000
0.0242	0.0161	0.00472	2003	98182
0.0254	0.0173	0.00532	2102	82500
0.0267	0.0186	0.00601	2204	78462
0.0281	0.0200	0.00686	2295	65000
0.0297	0.0216	0.00785	2397	63750
0.0317	0.0236	0.00924	2500	51500
0.0329	0.0248	0.01014	2550	41667
0.0344	0.0263	0.01133	2601	34000
0.0364	0.0283	0.01313	2635	17000
0.0386	0.0305	0.01554	2599	-16364
0.0402	0.0321	0.01771	2504	-59375

Test # 6 — Torque: 60

Displ	Normalized Disp.	Estim. Joint Displ.	Load	Slope: Load/ Displ
0.0059	0.0000	0.00000	102	
0.0078	0.0019	0.00129	201	52105
0.0088	0.0029	0.00166	303	102000
0.0094	0.0035	0.00168	396	155000
0.0101	0.0042	0.00171	502	151429
0.0108	0.0049	0.00173	614	160000
0.0113	0.0054	0.00175	698	168000
0.0119	0.0060	0.00178	802	173333
0.0126	0.0067	0.00180	895	132857
0.0134	0.0075	0.00193	1003	135000
0.0143	0.0084	0.00217	1110	118889
0.0149	0.0090	0.00223	1197	145000
0.0158	0.0099	0.00251	1297	111111
0.0166	0.0107	0.00272	1392	118750
0.0176	0.0117	0.00305	1501	109000
0.0186	0.0127	0.00340	1606	105000
0.0196	0.0137	0.00378	1706	100000
0.0206	0.0147	0.00420	1801	95000
0.0218	0.0159	0.00476	1903	85000
0.0229	0.0170	0.00526	2000	88182
0.0243	0.0184	0.00604	2101	72143
0.0258	0.0199	0.00692	2202	67333
0.0274	0.0215	0.00793	2296	58750
0.0288	0.0229	0.00898	2353	40714
0.0313	0.0254	0.01121	2397	17600
0.0346	0.0287	0.01513	2297	-30303
0.0361	0.0302	0.01720	2204	-62000
0.0379	0.0320	0.02023	2006	-110000

Test 7 — Torque 60

Displ	Normalized Disp.	Estim. Joint Displ.	Load	Slope: Load/ Displ
0.0099	0.0000	0.00000	104	
0.0116	0.0017	0.00099	203	58235
0.0127	0.0028	0.00136	306	93636
0.0134	0.0035	0.00145	391	121429
0.0141	0.0042	0.00148	489	140000
0.0147	0.0048	0.00150	592	171667
0.0156	0.0057	0.00153	702	122222
0.0164	0.0065	0.00155	810	135000
0.0171	0.0072	0.00157	894	120000
0.0179	0.0080	0.00166	994	125000
0.0187	0.0088	0.00176	1092	122500
0.0196	0.0097	0.00195	1191	110000
0.0206	0.0107	0.00213	1306	115000
0.0214	0.0115	0.00227	1399	116250
0.0224	0.0125	0.00250	1508	109000
0.0232	0.0133	0.00265	1598	112500
0.0242	0.0143	0.00291	1703	105000
0.0252	0.0153	0.00320	1802	99000
0.0264	0.0165	0.00366	1906	86667
0.0276	0.0177	0.00418	2001	79167
0.0289	0.0190	0.00481	2096	73077
0.0306	0.0207	0.00579	2197	59412
0.0326	0.0227	0.00707	2298	50500
0.0341	0.0242	0.00819	2351	35333
0.0359	0.0260	0.00981	2376	13889
0.0389	0.0290	0.01334	2301	-25000
0.0404	0.0305	0.01556	2200	-67333
0.0424	0.0325	0.01903	1994	-103000

Test 8 — Torque: 60

Displ	Normalized Disp.	Estim. Joint Displ.	Load	Slope: Load/ Displ
0.0044	0.0000	0.00000	104	
0.0059	0.0015	0.00104	195	60667
0.0069	0.0025	0.00155	292	97000
0.0078	0.0034	0.00188	404	124444
0.0084	0.0040	0.00191	514	183333
0.0089	0.0045	0.00193	605	182000
0.0094	0.0050	0.00195	700	190000
0.0099	0.0055	0.00197	797	194000
0.0104	0.0060	0.00200	896	198000
0.0109	0.0065	0.00202	994	196000
0.0114	0.0070	0.00204	1094	200000
0.0121	0.0077	0.00209	1214	171429
0.0126	0.0082	0.00218	1294	160000
0.0133	0.0089	0.00236	1397	147143
0.0139	0.0095	0.00247	1494	161667
0.0148	0.0104	0.00283	1602	120000
0.0156	0.0112	0.00315	1697	118750
0.0166	0.0122	0.00364	1797	100000
0.0178	0.0134	0.00430	1904	89167
0.0189	0.0145	0.00493	1998	85455
0.0203	0.0159	0.00584	2095	69286
0.0218	0.0174	0.00682	2198	68667
0.0236	0.0192	0.00808	2304	58889
0.0264	0.0220	0.01039	2400	34286
0.0276	0.0232	0.01154	2410	8333
0.0308	0.0264	0.01527	2305	-32813
0.0319	0.0275	0.01688	2206	-90000
0.0336	0.0292	0.01964	1996	-123529

Test 9 — Torque: 100

Displ	Normalized Disp.	Estim. Joint Displ.	Load	Slope: Load/ Displ
0.0084	0.0000	0.00000	102	
0.0101	0.0017	0.00112	207	61765
0.0109	0.0025	0.00146	292	106250
0.0117	0.0033	0.00164	405	141250
0.0124	0.0040	0.00166	510	150000
0.0129	0.0045	0.00168	594	168000
0.0136	0.0052	0.00171	716	174286
0.0141	0.0057	0.00173	812	192000
0.0146	0.0062	0.00175	912	200000
0.015	0.0066	0.00178	1016	260000
0.0154	0.0070	0.00179	1085	172500
0.0159	0.0075	0.00182	1188	206000
0.0166	0.0082	0.00185	1314	180000
0.0171	0.0087	0.00186	1396	164000
0.0177	0.0093	0.00189	1497	168333
0.0184	0.0100	0.00191	1592	135714
0.0192	0.0108	0.00203	1700	135000
0.0201	0.0117	0.00238	1799	110000
0.021	0.0126	0.00269	1907	120000
0.0221	0.0137	0.00325	2006	90000
0.023	0.0146	0.00364	2098	102222
0.0242	0.0158	0.00429	2199	84167
0.0255	0.0171	0.00500	2306	82308
0.0269	0.0185	0.00589	2398	65714
0.0287	0.0203	0.00712	2502	57778
0.0299	0.0215	0.00804	2554	43333
0.0312	0.0228	0.00909	2600	35385
0.0332	0.0248	0.01090	2633	16500
0.0351	0.0267	0.01298	2601	-16842
0.0367	0.0283	0.01511	2505	-60000

Test 10 — Torque: 100

Displ	Normalized Disp.	Estim. Joint Displ.	Load	Slope: Load/ Displ
0.0036	0.0000	0.00000	103	
0.0053	0.0017	0.00117	199	56471
0.0063	0.0027	0.00159	303	104000
0.0070	0.0034	0.00180	391	125714
0.0076	0.0040	0.00182	493	170000
0.0083	0.0047	0.00185	603	157143
0.0088	0.0052	0.00187	690	174000
0.0095	0.0059	0.00190	813	175714
0.0100	0.0064	0.00192	910	194000
0.0105	0.0069	0.00194	1009	198000
0.0110	0.0074	0.00197	1111	204000
0.0115	0.0079	0.00199	1215	208000
0.0120	0.0084	0.00201	1310	190000
0.0125	0.0089	0.00203	1395	170000
0.0131	0.0095	0.00206	1500	175000
0.0138	0.0102	0.00208	1596	137143
0.0146	0.0110	0.00210	1706	137500
0.0155	0.0119	0.00244	1808	113333
0.0163	0.0127	0.00271	1902	117500
0.0173	0.0137	0.00317	2001	99000
0.0185	0.0149	0.00380	2103	85000
0.0196	0.0160	0.00437	2199	87273
0.0210	0.0174	0.00522	2297	70000
0.0226	0.0190	0.00625	2400	64375
0.0236	0.0200	0.00697	2450	50000
0.0250	0.0214	0.00808	2502	37143
0.0278	0.0242	0.01061	2552	17857
0.0300	0.0264	0.01311	2498	-24545
0.0321	0.0285	0.01627	2307	-90952

Test 11 — Torque: 100

Displ	Normalized Disp.	Estim. Joint Displ.	Load	Slope: Load/ Displ
0.0046	0.0000	0.00000	103	
0.0066	0.0020	0.00139	203	50000
0.0078	0.0032	0.00195	306	85833
0.0086	0.0040	0.00220	395	111250
0.0095	0.0049	0.00245	501	117778
0.0101	0.0055	0.00247	595	156667
0.0108	0.0062	0.00250	697	145714
0.0115	0.0069	0.00252	806	155714
0.0120	0.0074	0.00254	893	174000
0.0126	0.0080	0.00257	1012	198333
0.0131	0.0085	0.00259	1100	176000
0.0136	0.0090	0.00261	1189	178000
0.0143	0.0097	0.00264	1293	148571
0.0151	0.0105	0.00266	1408	143750
0.0158	0.0112	0.00268	1499	130000
0.0166	0.0120	0.00275	1608	136250
0.0173	0.0127	0.00293	1692	120000
0.0183	0.0137	0.00321	1809	117000
0.0191	0.0145	0.00348	1895	107500
0.0203	0.0157	0.00402	2003	90000
0.0215	0.0169	0.00465	2096	77500
0.0230	0.0184	0.00548	2204	72000
0.0245	0.0199	0.00637	2304	66667
0.0261	0.0215	0.00737	2401	60625
0.0286	0.0240	0.00927	2499	39200
0.0305	0.0259	0.01102	2524	13158
0.0321	0.0275	0.01276	2500	-15000
0.0351	0.0305	0.01699	2300	-66667
0.0375	0.0329	0.02126	1997	-126250

Test 12 — Torque: 100

Displ	Normalized Disp.	Estim. Joint Displ.	Load	Slope: Load/ Displ
0.0086	0.0000	0.00000	104	
0.0102	0.0016	0.00113	198	58750
0.0112	0.0026	0.00157	310	112000
0.0118	0.0032	0.00167	410	166667
0.0123	0.0037	0.00169	493	166000
0.0130	0.0044	0.00171	611	168571
0.0135	0.0049	0.00174	708	194000
0.0140	0.0054	0.00176	813	210000
0.0143	0.0057	0.00178	887	246667
0.0148	0.0062	0.00180	1001	228000
0.0153	0.0067	0.00183	1116	230000
0.0157	0.0071	0.00185	1195	197500
0.0162	0.0076	0.00187	1309	228000
0.0167	0.0081	0.00190	1419	220000
0.0170	0.0084	0.00191	1490	236667
0.0175	0.0089	0.00194	1590	200000
0.0182	0.0096	0.00196	1697	152857
0.0188	0.0102	0.00198	1794	161667
0.0197	0.0111	0.00208	1903	121111
0.0205	0.0119	0.00239	2002	123750
0.0215	0.0129	0.00286	2108	106000
0.0225	0.0139	0.00339	2202	94000
0.0237	0.0151	0.00410	2299	80833
0.0250	0.0164	0.00492	2395	73846
0.0260	0.0174	0.00562	2454	59000
0.0272	0.0186	0.00657	2504	41667
0.0290	0.0204	0.00822	2534	16667
0.0307	0.0221	0.01008	2502	-18824
0.0333	0.0247	0.01373	2293	-80385

Estimated slope at full gap (from tests 1 - 4):
104036 in the range of 1000 to 1500 lb
96299 between 1600 and 1800 lb)
89038 at 1900 lb
81371 at 2000 lb
72132 at 2100 lb
65873 at 2200 lb
54749 at 2300 lb

Proportional limit	1800	1700	1700	1600	900	900	950	1200	1600	1650	1550	1800
Full gap estimated based on gapped slope					1700	1400	1700	1700	2300	2200	2000	2400
Estimated preload	0	0	0	0	1278.1955	1052.632	1278.195	1278.195	1840	1760	1600	1920
Torque factor, K					0.25	0.30	0.25	0.25	0.29	0.30	0.33	0.27

Ab (sq in)	0.020	Tensile stress area for bolt	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.020
Eb (psi)	29000000	Modulus of bolt material	29000000	29000000	29000000	29000000	29000000	29000000	29000000	29000000	29000000	29000000	29000000
Lb (in)	0.40	Effective length of bolt	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40
Kb (lb/in)	1450000	Bolt stiffness (= AE/L)	1450000	1450000	1450000	1450000	1450000	1450000	1450000	1450000	1450000	1450000	1450000
Kj (lb/in)	1450000	Assumed joint stiffness (bolt & fittings)				4350000	4350000	4350000	4350000	4350000	4350000	4350000	4350000
K (lb/in)	103101	Total measured stiffness in linear region	101159	101735	102165	160645	155938	135946	189231	174828	173000	156800	190800
Kf (lb/in)	110993	Derived stiffness of test fixture	108746	109411	109909	166805	161735	140332	197837	182148	180165	162663	199553
Kf (lb/in)	181132	Average fixture K from tests 9 - 12	181132	181132	181132	181132							
Kj (lb/in)	239326	Best guess at joint stiffness	229118	232090	234342	1420302							

Ult. Load:	2481	2495	2419	2318	2635	2397	2376	2410	2633	2552	2524	2534
Mean value:		145445.6										
Tests 1 - 4:	2428.25											
Tests 5 - 8:	2454.5											
Tests 9 - 12:	2560.75											
All tests:	2481.167											
	2782.563	4455.563	85.5625	12155.063	32580.25	3306.25	6162.25	1980.25	5220.063	76.5625	1350.563	715.5625
	0.027778	191.3611	3864.694	26623.361	23664.694	7084.028	11060.03	5064.694	23053.36	5017.361	1834.694	2791.361
Standard deviation:												
Tests 1 - 4:	80.57864											
Tests 5 - 8:	121.1459											
Tests 9 - 12:	49.54039											
All tests:	100.1134											

Average estimated bolt preload:
Tests 5 - 8: 1221.8045
Tests 9 - 12:

FalconSat-2

Structural Verification Report

Appendix G

Testing of the Ground Handling Interface

MEMO

To: Jerry Sellers

From: Ron Humble

CC: Tom Sarafin

Date: 20 May 2002/ Revised 23 May 2002

Subject: OSHA and GSFC Required Lifting Hardware Verification

OSHA requires that all lifting fixtures be verified for adequate strength prior to delivery to a user. GSFC, in turn, requires that all spacecraft delivered to them meet the OSHA standards.

Verification Requirement

My understanding of the OSHA/GSFC requirements is as follows:

1. The lifting hardware must be analyzed to show adequate strength to five-times the expected maximum lifting load without rupture.
2. The lifting hardware must be proof tested to two-times the expected maximum lifting load without rupturing.

The USAFA FalconSat-2 program has a further requirement that there can be no readily measurable or obvious permanent deformation after the proof test.

The FalconSat-2 spacecraft weighs-in at just a bit over 43 lbs based upon measurements made on the qualification unit and expected masses on the flight unit. Although we are not required by GSFC to lift our spacecraft and the PES hardware, it seems prudent to ensure that we can lift this hardware, if needed. From discussions with Mike Urban at GSFC, the PES hardware weighs-in at under 50 lbs. Therefore for the purposes of the OSHA lift-fixture verification, we will assume a maximum lifting load of 100 lbs and qualify the lifting hardware to this level.

A photo of the lifting lug configuration is shown in figure 1. The lug is mounted to the center of the top panel, as shown, with two ¾" 10-32 bolts passing through the top panel and threaded, with Spiralock inserts, into the lug itself.

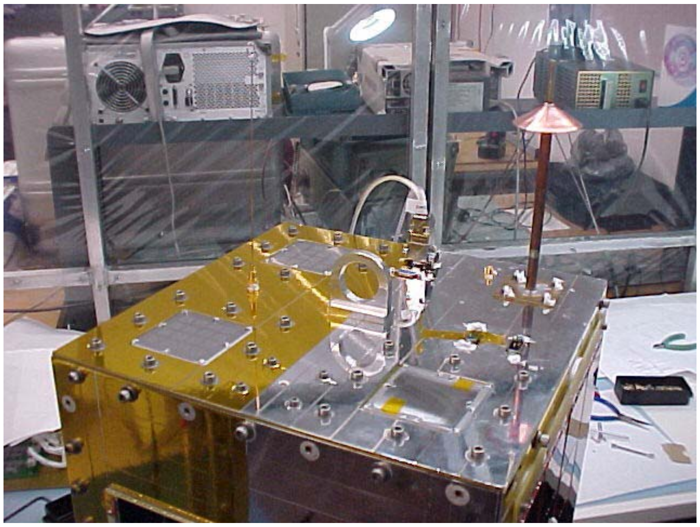


Figure 1, Installation of the Lifting Lug

Verification of the 5-Times Max Load Capability

To verify the strength of the lifting lug, we decided to perform a strength test of a test-lug to failure. This approach avoids questions and the uncertainties involved with an analytical approach. A lug was manufactured exactly as planned for the FalconSat-2 spacecraft and a special pull-test fixture was also constructed. The pull-test assembly is shown disassembled in figure 2 and assembled in figure 3, with a ½" diameter quick-release shear pin. The two cylindrical ends of the test assembly are interfaced with the pull-test machine with ¾" UNC bolts as shown in figure 4.

Figure 2, Disassembled Pull-Test Hardware

Figure 3, Assembled Pull-Test Hardware

Figure 4, Pull-Test Setup

Figure 5 shows the results of the pull test. The slope of the load versus deflection curve is quite linear up to over 1700 lbs indicating that the materials are still behaving somewhat elastically to loads more than three times the required 500 lb requirement. The test assembly finally failed at about 6200 lb giving a huge strength margin for this assembly. Figure 6 shows the assembly after failure. In fact, the pull-test adapter fitting ruptured before the lug failed.

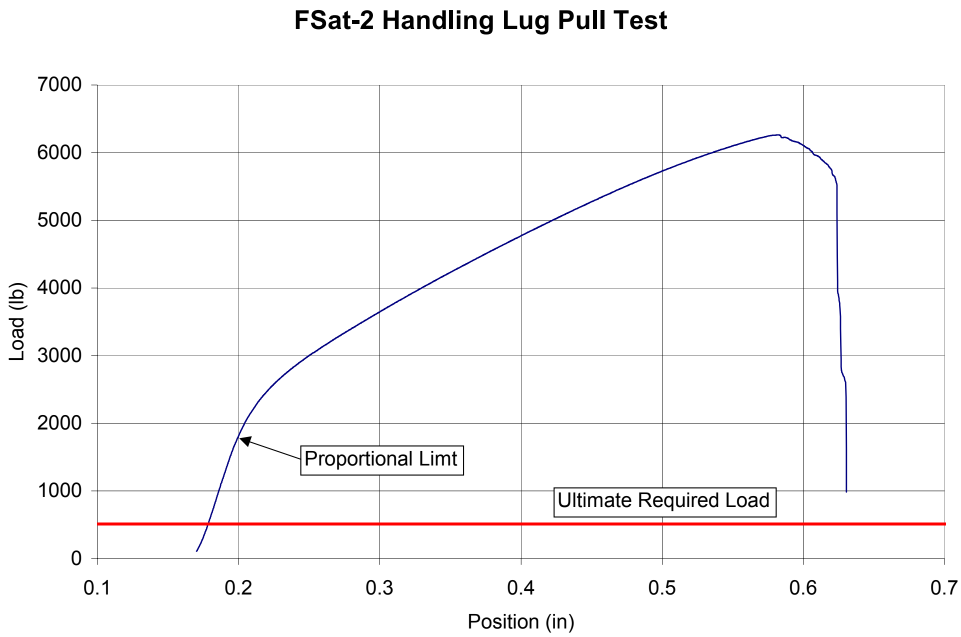


Figure 5, Load versus Relative Jaw Position for the Lifting Lug Pull Test

Figure 6, Pull Test Assembly After Rupture

Proof Test of the Two-Times Load Capability

To verify the lifting structure to two times the maximum load capability (2 x 100 lb). We assembled the FalconSat-2 flight structure as shown in figure 7. This assembly just included the primary structure and did not include any electronics or solar panel hardware. As such, the assembly weighed-in at approximately 26 lbs.

Figure 7, Spacecraft Undergoing 2-Times Lift Verification

To verify the strength of the lifting system, 4-45 lb weights were suspended from the base-ring of the spacecraft as shown in Figure 7. This brought the total suspended weight, on the lug, to:

$$26 + 4 \times 45 = 206 \text{ lbs}$$

which is 6 lbs above the 200 lb requirement.

During and after the test, a machinist's square was placed on the top of the spacecraft to measure deflection of the top panel. We estimate that the panel deflected elastically by about 0.1" under the full 206 lbs of load. However, after the suspended weights were removed, there was no detectable, residual deflection.

Conclusion

The FalconSat-2 spacecraft lifting hardware meets, and exceeds by a large margin, the lifting verification requirements of OSHA and NASA/GSFC.